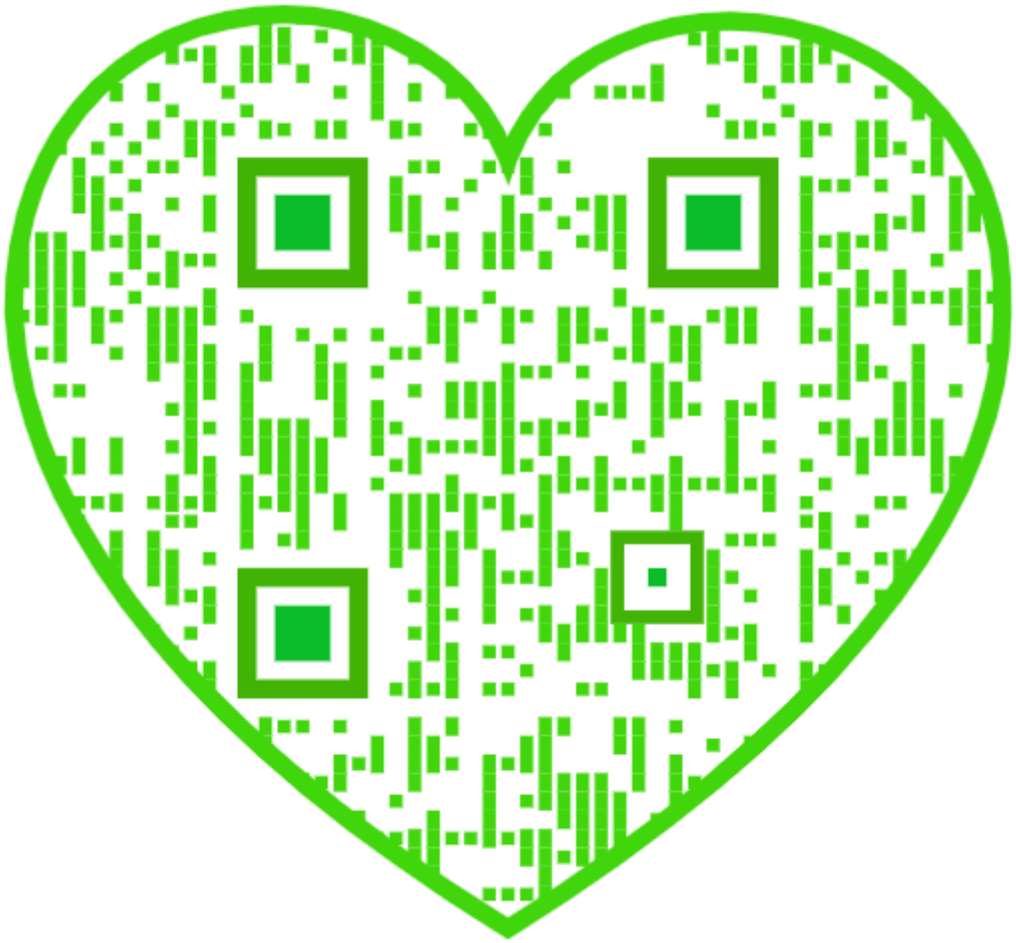


PUMPING MACHINERY.

A PRACTICAL HAND-BOOK

RELATING TO THE

CONSTRUCTION AND MANAGEMENT

OF

STEAM AND POWER PUMPING MACHINES.

BY

WILLIAM M. BARR,

MEMBER AMERICAN SOCIETY MECHANICAL ENGINEERS.

WITH UPWARDS OF TWO HUNDRED AND SIXTY ENGRAVINGS, COVERING EVERY ESSENTIAL DETAIL IN PUMP CONSTRUCTION.

PHILADELPHIA:

J. B. LIPPINCOTT COMPANY.

1893.

COPYRIGHT, 1893,
BY
WILLIAM M. BARR.

PRINTED BY J. B. LIPPINCOTT COMPANY, PHILADELPHIA, U. S. A.

29842

6504264

.C4P
.L27

PREFACE.

THE author has long felt the need, for his personal use, of a book similar to the one he now publishes, and had there been available a work covering approximately the same ground as the one now offered, the preparation of this volume would not have been undertaken. No apology is thought to be necessary for its appearance at this time, for, so far as I am aware, no other book similar in scope has been published in this country.

This book, as will be seen upon examination, is essentially descriptive of pump detail; no attempt has been made to enter into the theory and mathematics of pump-construction. The number of leading pump-makers in this country is comparatively few, and such are well equipped for the designing and building of pumping machinery; but there is another and larger body, comprising engineers, architects, contractors, plumbers, etc., who have occasion to recommend and use pumping machinery, and who wish to inform themselves regarding pump construction,—it is for the benefit of this latter class that this volume has been prepared. This book being largely descriptive, its illustrations have been made a prominent feature, the writer regarding a single suggestive sketch of more real value than a page of reading-matter.

With few exceptions, the illustrations are from pumping machinery actually constructed and in use, and those familiar with the author's work will not fail to recognize how large a proportion of the whole has been transferred directly from his own practice. Inasmuch as a large portion of this book has been in constant use by the writer for years, it is believed that in its present convenient form it will prove valuable, or at least suggestive, to others also. In addition to my own experience,

I have been particularly fortunate in having the friendly advice and co-operation of other engineers well known as successful designers of pumping machinery; these have contributed much valuable information, accompanied by drawings, here reproduced. Interesting and valuable extracts have also been made from foreign publications not generally accessible except in the larger cities.

Chapters XIII. and XVII. were originally intended to be added to this volume as an appendix, but are now incorporated with it for the following reasons:

The specifications of the Underwriters' pump are so valuable and complete that they ought not to be omitted, and to this end the chapter on duplex pumps was purposely abridged that the two might not parallel each other. The design and construction of duplex pumps has engaged almost the whole attention of the writer for many years, and it was with some hesitancy that he relinquished his own subject-matter for the insertion of what might be regarded as an Underwriters' circular; however that may be, the fact remains that it is a valuable contribution to the literature of the subject, and well worth the space given it.

The first part of the chapter on duty-trials is an abstract of an accepted report of the Duty-Trial Committee of the American Society of Mechanical Engineers, replacing my rewritten lecture on that subject delivered before the Franklin Institute. The substitution of the committee's report for my own subject-matter is in deference to the excellence of the committee's work, and to contribute to the earnest desire that for the sake of comparison and uniformity of records it should receive the support of the Society's membership.

The merit of a hand-book like this consists largely in the judicious selection and arrangement of its contents rather than upon strict originality, so that whatever selections have been useful to the writer he now transfers to the reader with the added results of his experience.

WILLIAM M. BARR.

PHILADELPHIA, January, 1893.

CONTENTS.

CHAPTER		PAGE
I.	INTRODUCTION	7
II.	WATER-PISTONS AND PLUNGERS	15
III.	PISTON- AND PLUNGER-RODS	37
IV.	WATER-VALVES AND SEATS	47
V.	AIR- AND VACUUM-CHAMBERS	90
VI.	SUCTION- AND DELIVERY-PIPES	98
VII.	WATER-END DESIGN	126
VIII.	HYDRAULIC-PRESSURE PUMPS	152
IX.	STEAM AND POWER CRANK-PUMPS	173
X.	DIRECT-ACTING STEAM-PUMPS	200
XI.	THE DUPLEX PUMP	226
XII.	COMPOUND DIRECT-ACTING STEAM-PUMPS	235
XIII.	FIRE-PUMPS	271
XIV.	MINING-PUMPS	289
XV.	ROTARY PUMPS	325
XVI.	CENTRIFUGAL PUMPS	334
XVII.	DUTY-TRIALS OF PUMPING ENGINES	361
XVIII.	HIGH-DUTY PUMPING ENGINES—DIRECT-ACTING	387
XIX.	HIGH-DUTY PUMPING ENGINES—FLY-WHEEL	411

PUMPING MACHINERY.

CHAPTER I.

INTRODUCTION.

THE art of raising water must of necessity have been one of the first of the mechanic arts to engage the attention of man, for no progress in civilization can be had without a convenient and ample supply of good and wholesome water. The earliest water-supply must have been the permanent springs and water-courses; but the growth of population, the increase in wealth, and a higher civilization required a broader development of the land, and it was during this period of development that the ingenuity of man was exercised in originating schemes and appliances for the lifting and distribution of water; but of the origin and early history of this art we know nothing.

It is not probable that any satisfactory device for raising water is lost to us, although its history has long since been forgotten, the fact of utility has been the very means of its preservation; nor did the great invasions and conquests of the ancient world affect unfavorably the development of this useful art, for water was alike essential to the conqueror and the conquered.

Machines for raising water admit of a great variety of forms, all depending upon the conditions of supply and delivery. The sources of supply are usually streams and wells; the ordinary delivery ranges from the simple lifting for irrigation to that of high service water-supply for cities.

A water-elevator has been defined (Knight) as a device for raising buckets from wells, and a pump as a device for lifting water by the motion of a piston in a cylinder.

Among the more important water-elevators are the following:

Archimedian screw,
Baling-machine,
Bascule,
Bucket-wheel,
Chapelet,
Dutch scoop,
Ejector,
Flash-wheel,
Flush-wheel,
Hydraulic belt,
Hydraulic ram,
Jantu,
Mental,
Noria,
Persian-wheel,
Picotah,
Scoop,
Scoop-wheel,
Shadûf,
Swape,
Turbine,
Tympanum,
Water-screw.

A partial list of pumps will include the following:

Bellows-pump,
Centrifugal,
Chain,
Chapelet,
Diaphragm-plunger,
Draining-pump,
Eccentric,
Ejector,
Elastic-piston,
Hydrapult,
Injector,
Pendulum,
Piston,
Plunger,
Rope,
Rotary,
Spiral,
Steam jet pump,
Steam-vacuum pump,
Syringe,
Vacuum,
Water-ram,
Water-screw,
Water-snail.

By reason of the limited scope of the present work it will be impossible to illustrate and describe so formidable an array of water-raising machines included in the partial lists given above. The reader is referred to Knight's "Mechanical Dictionary" for definitions, and especially to Ewbank's "Hy-

draulics" for illustrations, description, and history of early and curious water-raising devices.

Atmospheric Pressure.—It must not be inferred that the ancients were unacquainted with the physical properties of the atmosphere, and that they did not take it into account in the development of their hydraulic machines. There is every reason to believe that they understood and applied certain principles relating to the atmosphere; for example, the ancient Egyptians understood and used the siphon at least fourteen hundred and fifty years before the Christian era, which clearly indicates that they were acquainted with some facts regarding the expansibility, as well as the compressibility, of the air; but this was only a partial knowledge, for it is not clear that the exact data regarding atmospheric pressure were known until the middle of the seventeenth century. So also the suspension of a liquid in inverted vessels by the atmosphere, such as the atmospheric sprinkling-pot, was known in the earliest historic times, or, at least, was well known in the fifteenth century B.C.

The Syringe.—Few ancient devices could be pointed out that have given rise to more important improvements in the arts than the primitive syringe. Its modifications exert an extensive and beneficial influence in society. As a piston-bellows it is still extensively used in Oriental smitheries. It may be .considered as the immediate parent of the forcing if not of the atmospheric pump, in both of which it has greatly increased the comforts and conveniences of civilized life.

Suction is a word which has come down to us from a vast antiquity. The operation of sucking, as in the case of an infant, the sucking of poison from a wounded part by the application of the lips, are well-known illustrations. So also the raising of a liquid through a tube into the mouth. This operation has long been known as suction, and it was formerly

believed that it was effected by some power or faculty of the mouth independently of any other influence.

Suction is simply a term used to denote the absence or the removal of the atmosphere, so as to permit the flow of the liquid; suction does not raise the liquid, nor does it help to raise it. The term sucker for the valve attached to the pump-rod in an ordinary lift-pump no doubt had its origin in the fancied similarity of its action as compared with that of the mouth.

An atmospheric pump is merely a contrivance placed at the upper end of a pipe to remove the pressure of the atmosphere there, while it is left free to act on the liquid in which the lower end is immersed. It is immaterial what the substance of the machine is, or what figure it is made to assume, for any device by which air can be removed from the interior of a vessel is or may be used as a pump to raise water; there will be required, however, two valves, one opening upwards and placed in any part of the pipe or in the machine itself, to allow the water to pass up through it, but none to descend; the other valve placed over an aperture opening outwards, through which the contents of the vessel can be discharged, and at the same time prevent the entrance of external air. Just how long it took the earlier inventors to determine the "limit of suction" is not known, but the exact weight or pressure of the atmosphere was not authoritatively announced until after the experiments of Torricelli, in 1608, and subsequently confirmed by Pascal forty years later. The fact was then fully established that an atmospheric pump must be placed within twenty-six or twenty-eight feet of the surface of the water to be lifted; but, owing to the difficulty in getting tight joints in the suction-pipe, this distance was gradually shortened until twenty-two to twenty-five feet was regarded as the practical or ordinary limit of suction.

Ewbank records a singular incident of a tinman of Seville, who undertook to raise water from a well sixty feet deep by a common pump. Instead of making the sucker play within

thirty feet of the water, he made the rod so short that it did not reach within fifty feet of it. As a necessary consequence he could not raise any. Being greatly disappointed, he descended the well to examine the pipe, while a person above was employed in working the pump; and at last, in a fit of despair at his want of success, he dashed the hatchet or hammer in his hand violently against the pipe. By this act a small opening was made in the pipe about ten feet above the water, when, what must have been his surprise! the water instantly ascended, and was discharged at the spout.

This fact being published (1776) led to a reinvestigation of the subject, and instead of overthrowing the received doctrine of atmospheric pressure, more fully confirmed it. It was ascertained that the air on entering the pipe became mixed with water, and which, therefore, instead of being carried up in an unbroken column, was raised in disjointed portions, or in the form of thick rain. This mixture being much lighter than water alone, a longer column of it could be supported by the atmosphere; and by proportioning the quantity of air admitted, a column of the compound fluid may be elevated one hundred or two hundred feet by the atmospheric pump.

CLASSIFICATION OF PUMPS.

The easy and natural classification of pumps would be to divide them into three classes:

I. Lift-pumps,
II. Force-pumps,
III. Lift- and force-pumps,

} reciprocating or rotary.

These may again be sub-classified into—

Single-acting pumps,
Double-acting pumps.

And still further into—

Vertical pumps,
Horizontal pumps.

If pumps be classified according to their details of construction, the list would be still further extended into—

Bucket-pumps,
Piston-pumps,
Plunger-pumps,
Bucket- and plunger-pumps,
Bucket- and piston-pumps,
Piston- and plunger-pumps (known as the differential plunger-pump),
Rotary pumps,
Centrifugal pumps.

These names indicate a particular form of construction, and not a new or distinct classification, for each of these latter pumps must necessarily be included in the former. This latter classification is a convenient one, and has been adopted by the writer for his present use. It may be said that it is not an exact or scientific arrangement,—this much is admitted at the outset,—but it is the commercial one, and, therefore, in the direct line of every-day use.

The increasing subdivision in business enterprises, and the growing importance of pumping machinery as a part of the plant, would seem to call for another classification of pumps adapted for special uses ; for example, acids, alkalies, ammonia, beer, bilge-water, bleacheries, breweries, dye-works, drainage, fire-pumps, gas-works, etc. A mere catalogue of names, with suggestions regarding suitable pumping machinery for each, would occupy more space than could be given it in the present work, and it is doubtful even then if such a presentation would prove satisfactory because of the repetitions which must inevitably occur.

Pumps for General Service.—There is no subject in which it is so difficult to give advice in a general way as in pumping machinery, because each pumping plant has its own special peculiarities which must be considered, and which may not apply to any other pumping plant. There are two things,

however, which come within ordinary practice, and if designs be made to accord with either or both, the greater part of pump-service will have been fully met.

The first one is, that if water-ends be made sufficiently strong to handle water at one hundred and fifty pounds pressure, fully eight-tenths of the ordinary run of pump requirements can be supplied. It is the common practice in designing water-ends for trade pumps to make the details of suitable size and form for this pressure. This will cover the highest fire-pressure, which is usually the severest test to which an ordinary trade pump is put. For small water-works the pressures rarely ever reach the one-hundred-and-fifty-pound limit, even when on direct service. For hydraulic elevator service the pressure seldom exceeds one hundred pounds per square inch, except in steel-works and other places where there is a general hydraulic system using very high pressures.

Tank-service usually calls for lighter pressures, ranging from twenty-five to fifty pounds, but it is not customary among steam-pump-makers to make any difference in the weight of the water-end; the size remaining the same, will require the same detail and workmanship, so that nothing but a small amount of cast iron would be saved, and that is not worth the cost of altering or making new core-boxes.

The second relates to steam-pressure, which does not in ordinary practice exceed eighty or ninety pounds, so that if a steam-end of a pump be designed for one hundred pounds pressure, factory, water-works, and other service will be amply provided for.

Combinations of such steam- and water-ends will, therefore, meet almost every requirement in ordinary hydraulic operations.

Pumps for special service for higher steam- and water-pressures, such as doubling either or both of them, will require new proportions. An increased steam-pressure will, in general, require nothing more than thicker castings and stronger bolting, the size of the ports, the distribution of

steam, and other details remaining much the same. For the water-end it is often best to entirely change its form, and this usually occurs. Coupled with this higher water-pressure is, with few exceptions, a smaller quantity of water to be delivered. This is especially so in the case of hydraulic-pressure pumps, but is not generally true of pumping for mines.

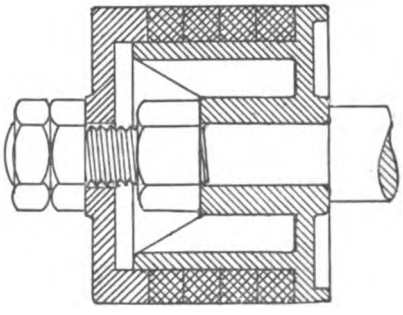
Whatever the service, pumps must be able to work continuously without danger of breakage, and this with as little expenditure of power as possible. But there are other considerations than the mere saving of coal: a pump must be simple, easily managed, and certain in its operation, or it will fail to meet the requirements of its owner, who seldom knows anything about pumping machinery. The very large business now annually done in direct-acting single and duplex pumps can probably be traced directly to their meeting the above conditions.

CHAPTER II.

WATER-PISTONS AND PLUNGERS.

Pistons.—The piston shown in section in Fig. 1 is the one in most common use. It consists of an iron or brass casting bored to fit the piston-rod, and turned on the outer flange to a loose fit in the bore of the cylinder; and also turned to a diameter suitable to the thickness of the packing to be used. The length of the piston should be such as to admit not less than three, or, better still, four rings of packing; to the end of the piston should be fitted a follower-plate and ring for compressing the packing. In the above illustration, which is suited to pistons of small and medium diameter, say eight inches or less, the piston-rod is turned down to allow a shoulder for driving the piston, and supplied with a nut for holding the piston firmly in position; a second nut, re-enforced by a jamb-nut, permits an adjustment of the follower and packing. Each follower should be fitted with two tapped holes for screwing in eye-bolts, to facilitate removal when the piston requires repacking.

FIG. 1.

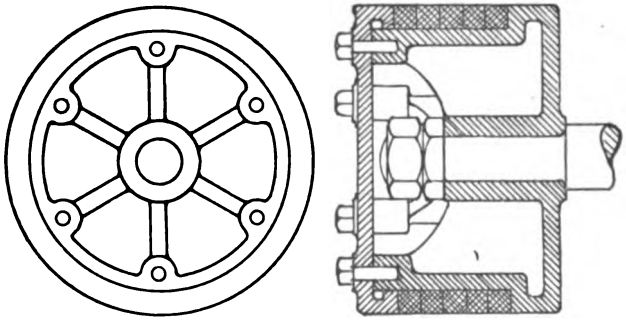


For larger pistons the design, Fig. 2, is used; it is in all respects the same as the former, except in the method of securing the follower.

The packing may be of any one of the numerous kinds now

offered to the trade, such as square-plaited flax, hemp, cotton with rubber core, etc. Another variety, made up of layers of

FIG. 2.



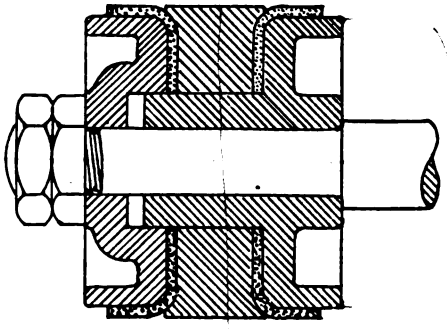
cotton-cloth and sheet-rubber, known to the trade as Tuck's packing, can be had in great variety in both width and thickness, but usually square, as ½″ × ½″, ⅝″ × ⅝″, ¾″ × ¾″, etc.

In packing a water-piston a word of caution may not be out of place here, and this caution applies particularly to the use of Tuck's packing, and all other packings of similar construction, and that is,—the packing must be cut not less than one width shorter than will permit the ends touching when wrapped around the piston; for example, ⅝″ packing should be cut ⅝″ shorter than the circumference of the outer rim of the piston, ¾″ packing to be cut ¾″ shorter, and in like manner for any other size. The reason for shortening the packing is that when new it is perfectly dry; but as it immediately absorbs moisture in the pump, it will soon swell tight enough to wholly prevent the movement of the piston if the ends of the packing are allowed to touch each other; by shortening the packing an end movement is permitted, instead of compelling a radial one.

Cup-leather-packed pistons are largely in use, especially for small pumps. Fig. 3 represents a sectional elevation

of such a piston; it is made up of three parts,—the piston-head, or that portion secured to the rod; the chunk-ring, or central-distance piece, and the follower; together with the two cup-leathers, as shown. Inasmuch as this type of piston does not require an adjustable follower, the piston-rod extends clear through, and fitted with a nut and jamb-nut, as shown.

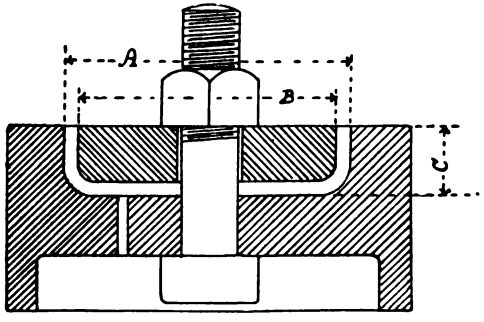
FIG. 3.



The leather must be of a solid oak-tanned quality, without soft spots or spongy places, and must be uniform in thickness; $\frac{1}{4}$ and $\frac{5}{16}$ of an inch being the ordinary or average thickness in selected hides. Nearly all leather dealers, especially those in the larger cities, have a specially-prepared leather for hydraulic work, at a price but little in advance over first-quality sole-leather.

A mould for making cup-leathers is shown in Fig. 4. The outside diameter, *A*, corresponds to that of the cylinder it

FIG. 4.

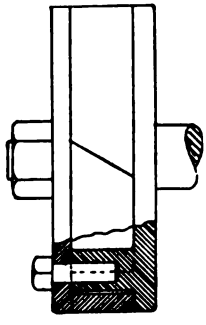


is intended to fit. The width of the groove must be suited to the thickness of the leather, together with a slight reduction

on the inside, *B*, say $\frac{1}{32}$ of an inch; the corners must have a radius sufficiently large, say ½ to ¾ of an inch, to prevent injuring the leather; the depth of the cup, *C*, should not be more than sufficient to make a tight joint. Leathers for a four-inch piston need not be cut more than 1¼ inches larger in diameter, and those for an eight-inch pump need not be more than 1⅝ inches larger. As the wear is confined to that portion of the cup-leather which touches the bore of the water-cylinder, there is no advantage in increasing its depth over the size given above.

To prepare a Set of Leathers.—After cutting out the disk of leather, with a central hole for the bolt to pass through, and making sure of its uniform thickness, it is then soaked in water until quite soft and pliable, after which it is placed in the mould with the grain or hair side down, so that that shall be the working side. The central part of the mould must now be very gradually tightened until the leather is in firm contact between the two faces of the mould. There will be more or less of a ragged edge above the top of the mould, which can be trimmed off with a knife or chisel. After the leather has taken a permanent set, as by remaining over-night in the mould, it may then be removed, after which it should be well greased with tallow, and is then ready for use. As cup-leathers thus pressed will permanently retain their shape, they may be made up in advance of requirements.

FIG. 5.



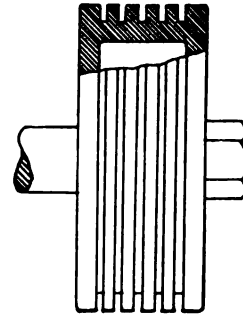
Pistons with Metal Rings.—It is not a common practice in this country to fit water-pistons with metal rings; but if for any reason such a packing is thought to be more desirable than those just described, a piston made with a single broad ring, as shown in Fig. 5, will be found quite satisfactory. This ring must be turned slightly larger than the bore of the cylinder in which it is to work, say ⅛ of an inch for

a piston twelve inches in diameter. The ring is to be split diagonally and a piece taken out the same as for steam-pistons; the ring should touch on the inclined edges when sprung down to its proper diameter. If made of cast iron or hard brass, the ring will be sufficiently elastic to fit the barrel of the pump until either or both are worn out.

There is a strong prejudice in this country against such a piston-packing for water, and in consequence pistons of this construction are not plentiful, the preference being given to those adapted for the use of fibrous packing.

A solid piston with grooves, as shown in Fig. 6, is not largely in use in this country; but it is a kind of piston which possesses some merit. It is simply a plain piston turned to fit the cylinder in which it is intended to work, and after finishing to size several grooves are turned to a moderate depth, say $\frac{3}{8}$ to $\frac{1}{2}$ of an inch on a twelve-inch piston; the width of the grooves may be about $\frac{1}{4}$ to $\frac{5}{16}$ of an inch.

FIG. 6.

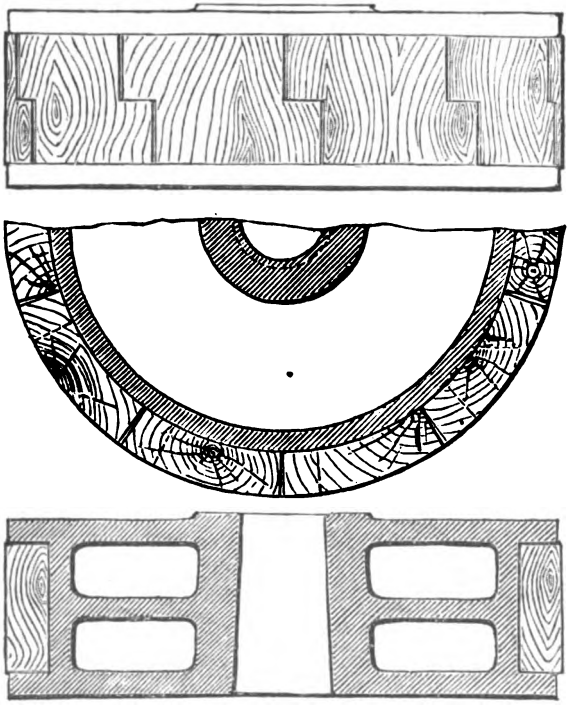


The efficiency of such a piston depends on the fact that sudden enlargements in any cavity through which water is to pass under pressure induces certain currents in the cavity which seriously impede, if they do not prevent, a direct passage of water. Now, in the case of a rapidly-moving piston this interference, as would be the case where several grooves occur, would nearly, if not entirely, prevent a flow of water in any direction before the reversal of the piston at the end of its stroke, at which time the direction of pressure would also be reversed, the result being a tight piston, water-packed.

The piston fitted with wood, shown in Fig. 7, is reproduced from Burgh's "Condensation of Steam." This piston was introduced in 1860 for use in circulating pumps for marine

engines. It is a metal-grooved disk, with blocks of wood (lignum-vitæ) fitted in the groove. The only feature worthy

FIG. 7.

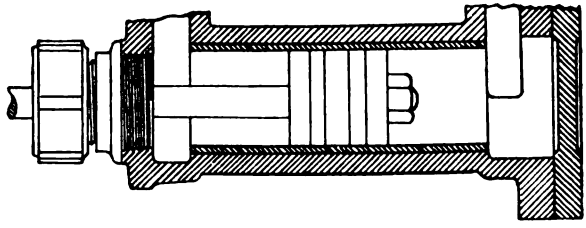


of comment is the fitting of the blocks with each other, which is shown in the top view.

Linings for Piston-Pumps.—In designing a water-end for a piston-pump it is considered good practice, but by no means a universal one, to line the barrel with a brass bushing. For small pumps, say less than six inches in diameter of piston, a seamless drawn tube pressed into a bored hole, and slightly upset or riveted over each end, as in Fig. 8, or made long enough to touch the heads, as in Fig. 9, is practised by pump-

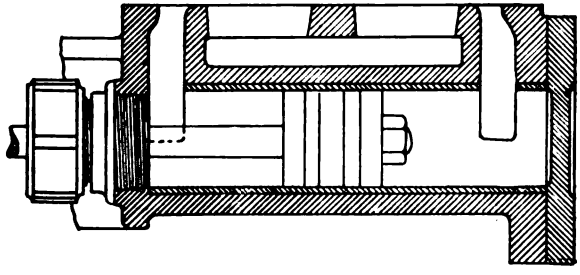
makers. The second is much the best arrangement of the two, because the lining can be more easily removed and a new one substituted; the objection to it, if any, being mainly one of first cost in manufacture, there being a longer hole to bore

FIG. 8.



in the pump-barrel and a longer tube to supply, which means more weight of brass; in addition to which is the cost of cutting a port through each end of the lining for the water. The objection to the first of the two examples is the great difficulty in supplying a distant customer with a tube which shall exactly fit the cylinder of his pump. The difference between a tight

FIG. 9.

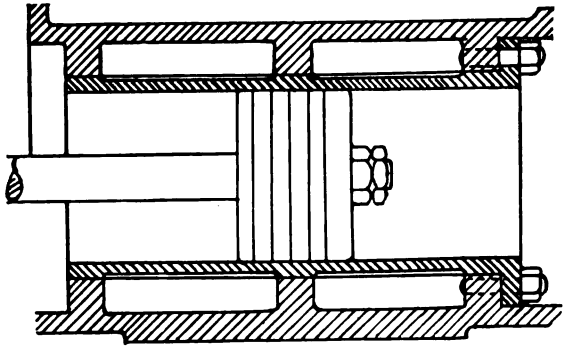


fit and a loose fit is here made more apparent than almost any other portion of the pump; for example, if the tube be $\frac{1}{1000}$ of an inch over size, it will be almost, if not quite, impossible to get it in place without special appliances; on the other hand, should it be $\frac{1}{1000}$ of an inch less in diameter than the bore of the water-cylinder, it will not fit, and cannot be used.

So small a dimension as that given above, when taken in connection with diameters of four to six inches, is almost impossible to register and make duplicate work to fit, except the pump be returned to the shop where it was built, and where are special appliances for forcing in the larger tube. If, however, the tube extend from end to end, as in Fig. 9, the smaller tube may be used, the heads preventing end-motion, and a film of water will make the tube tight in its place.

A removable lining, as shown in Fig. 10, is a better form than either of the above. In this case the barrel of the pump

FIG. 10.



is bored and the lining turned to fit. This lining is provided with a flange for securing it in place, a strong and well-designed arrangement. Incidentally, another advantage is had in the fact that the flange being circular admits of a regular spacing of bolt-holes, so that, in the event of the bottom of the bore of the lining becoming worn by the action of the piston, accompanied by sand or grit, and thus become scored or worn out, it is only necessary to smooth down the rough ridges and partially turn the lining in place so as to bring a new and unworn surface at the bottom, the fibrous packing adapting itself to the uneven or irregular circumference. These linings when worn out can readily be replaced by new ones.

BUCKET-PUMPS.

A bucket-pump may be described as a vertical, single-acting piston-pump, with one or more valves fitted to the top of the piston, opening upwards. These valves may be of any one of several varieties best suited to the work. For cistern-pumps, and usually for all pumps of small diameter and intended only for low lifts, leather valves weighted on the back,

FIG. 11.

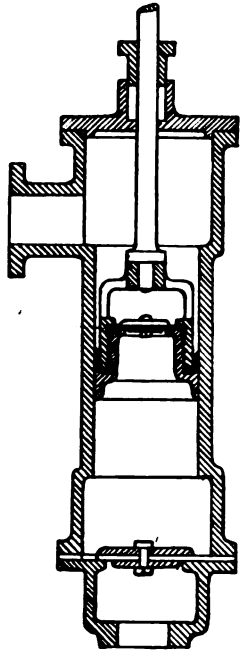
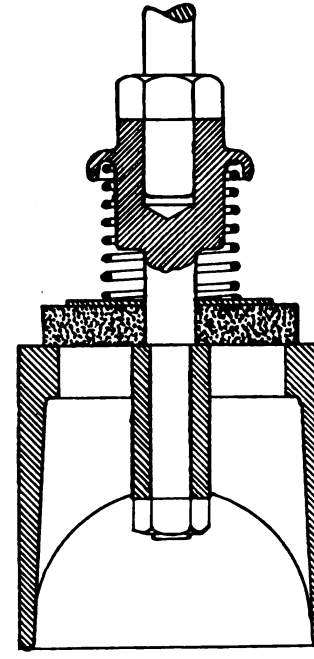


FIG. 12.



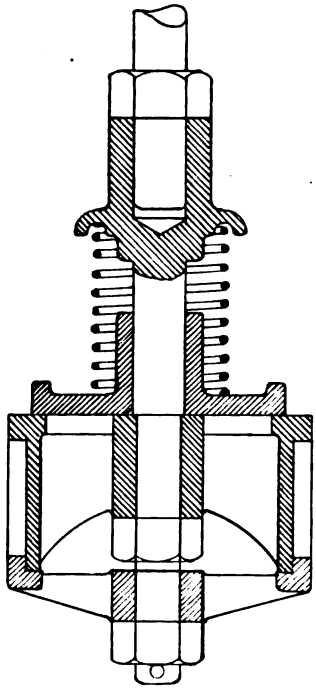
as illustrated in Fig. 11, are in general use, and give but little trouble, as wet leather makes an excellent hinge.

If a bucket-pump is to lift the water to a considerable elevation, say one hundred feet, and the diameter of the pump-barrel is sufficiently large to permit the use of a rubber valve, as shown in Fig. 12, satisfactory results are usually had. The valve-seat is made with four or more grids about ¼ of

an inch wide; the clear opening for the passage of water through the seat between the grids may be of any convenient shape, but should not present a larger area than two square inches, and as much less as circumstances will permit. The rubber disk may be ½ of an inch thick for diameters not exceeding three inches, and ¾ of an inch thick for valves six inches in diameter. A brass plate, $\frac{1}{16}$ of an inch thick and nearly the diameter of the valve, should be added to the top of the valve to distribute the pressure of the spiral spring over a larger area, and to prevent the spring wearing a circular groove in the back of the valve.

A pump-bucket for high pressure is shown in Fig. 13. The valve and seat are made of hard gun-metal, and designed for continuous service under one hundred and fifty pounds pressure. The valve and seat are flat, and will require to be carefully scraped to each other; the best way is to scrape the valve true to a surface-plate, and then scrape the seat to the valve. The engraving does not show it, but it is the common practice when facing the valve in a lathe to hollow out that portion of the valve between the outer and inner bearings to a distance of say $\frac{1}{32}$ of an inch below the face; this is to make the fitting easier, as only the surfaces in contact have then to be dealt with.

FIG. 13.



The bucket type of pump is used exclusively in drilled and oil-well service, and often at great depths. A variety of buckets and valves have been designed

for this service, but none have been so entirely satisfactory as those fitted with ball-valves and cup-leathers, as in Fig. 14. It is the practice to make these buckets and valves of the toughest gun-metal only; and as they are sometimes required to work under pressures which range from four

FIG. 14.

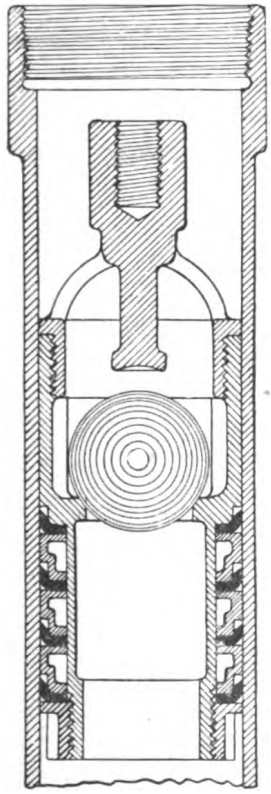
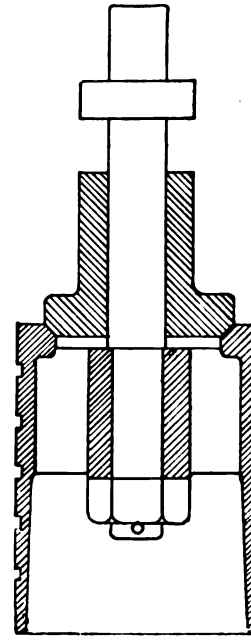


FIG. 15.



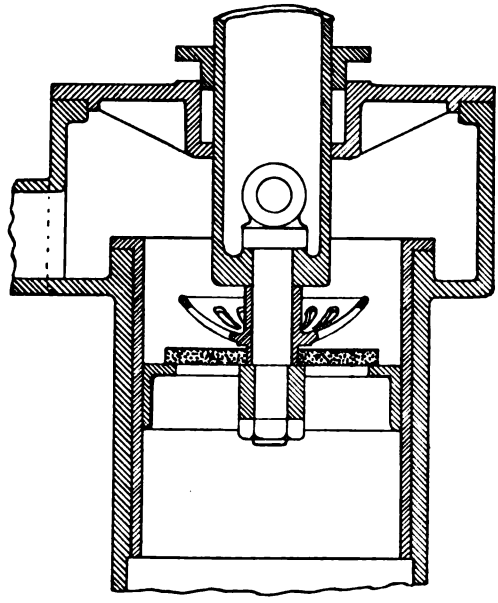
hundred to eight hundred pounds per square inch, the greatest care must be exercised in designing, that a proper valve area and strength of parts are secured.

Instead of using a ball-valve, an adaptation of the miter-valve, as shown in Fig. 15, may be used. This is an excellent form of valve for clear water, but will not work so well if there be much sand or grit in the water, as it gets between the valve and its spindle, and often prevents proper seating. It is for this reason that the ball-valve is given the preference for deep-

well pumps. The engraving includes a design for a water-packed bucket or piston, a style not much in use because of the liability of occurrence of sand in suspension in the water at the bottom of the well; it does not work satisfactorily except in clear water.

Air-Pump Bucket-Piston.—The bucket-pump is generally liked for both air and circulating pumps in vertical marine engines. Fig. 16 represents a type of bucket in very general use

FIG. 16.

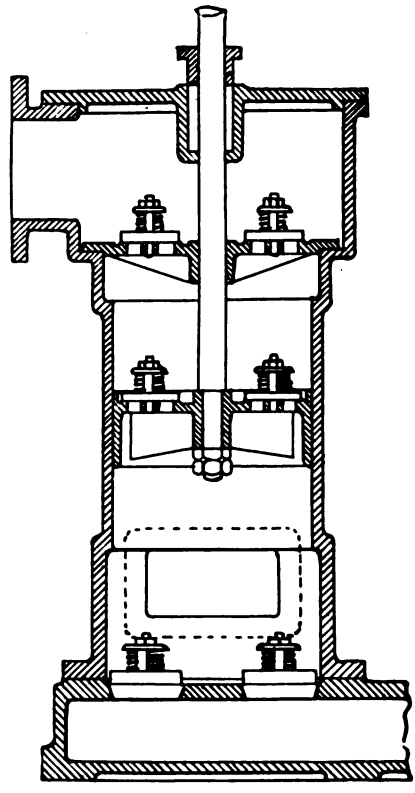


until within a few years. The bucket was usually, though not always, arranged for the use of fibrous packing; the top of the piston was made with suitable openings for the passage of the water through it; over these openings was placed a single rubber disk, and on the top of that a curved guard, a central bolt securing all together. The bucket descending into the water forced the rubber disk off the seat at its outer

edge only, it being firmly secured at the centre; this required of the valve a somewhat complicated series of internal movements at each stroke of the bucket, the result being destructive, as well as requiring considerable time for the recovery of the shape and proper seating of the valve.

The more recently-designed marine engines run at much higher speeds than was the practice ten years ago, and among the other changes in detail was that of the air and circulating pump-buckets; so that it is now the common practice to put in a number of small valves instead of one large one. (See Fig. 17.)

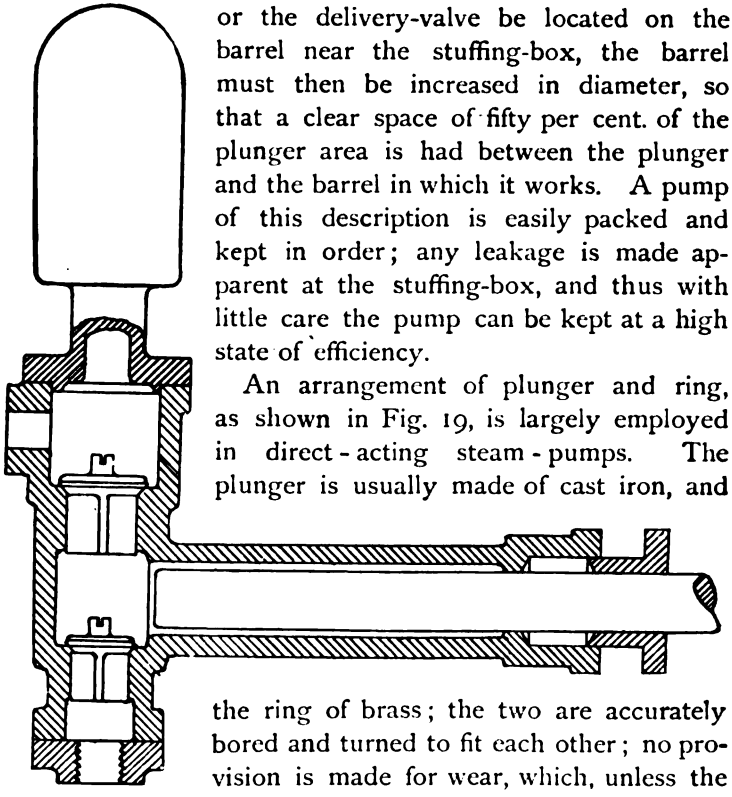
FIG. 17.



PLUNGER-PUMPS.

A plunger-pump is one in which a turned plunger passes through a stuffing-box, ring, or barrel of a pump, so as to alternately produce a vacuum for the water to flow into the pump and a pressure when forcing the water out of the pump. Fig. 18 is an illustration of a design of pump much used for feeding steam-boilers; the lower is the suction-valve, and the upper the delivery-valve. The plunger need not fit the barrel except at the stuffing-box. This pump is single-acting only. It is not necessary that the valves be one exactly above the

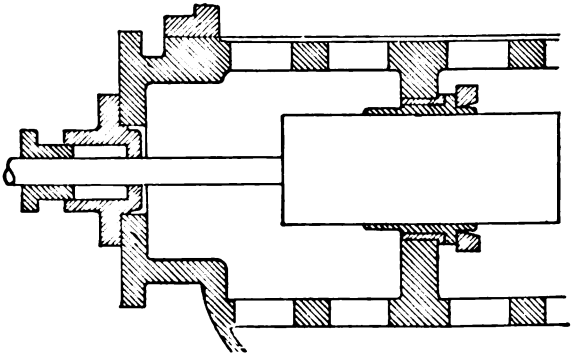
FIG. 18.



other, as shown; but if the suction-valve or the delivery-valve be located on the barrel near the stuffing-box, the barrel must then be increased in diameter, so that a clear space of fifty per cent. of the plunger area is had between the plunger and the barrel in which it works. A pump of this description is easily packed and kept in order; any leakage is made apparent at the stuffing-box, and thus with little care the pump can be kept at a high state of efficiency.

An arrangement of plunger and ring, as shown in Fig. 19, is largely employed in direct-acting steam-pumps. The plunger is usually made of cast iron, and the ring of brass; the two are accurately bored and turned to fit each other; no provision is made for wear, which, unless the water is gritty, is very slight. At first thought one would be

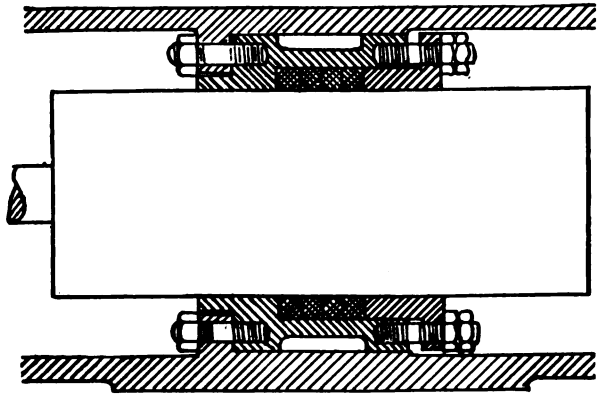
FIG. 19.



inclined to condemn this arrangement as faulty in design; but many years of service, covering thousands of steam-pumps, has completely demonstrated its excellence as a pump detail, when used in handling clear water, free from gritty matter.

Inside-Packed Plunger-Pump.—If the water to be pumped is gritty it is likely to bring extraordinary wear upon both the plunger and the ring when made solid. To obviate this an internal stuffing-box, as shown in Fig. 20, can be used

FIG. 20.



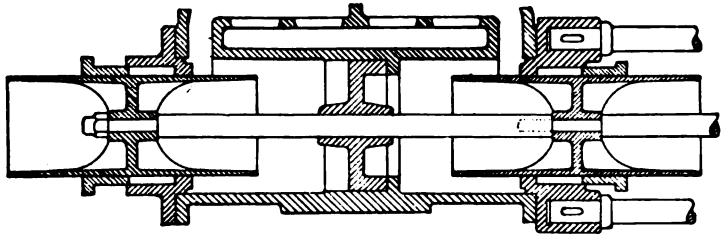
with advantage. This form of plunger-pump is to be preferred over a piston-pump for gritty water, inasmuch as the cost of a new plunger is much less than that of a new lining. There is a further advantage in the fact that slight reductions can be made in the diameter of the plunger by turning in a lathe to remove the scoring incident to the service in which it may be employed, the difference in diameter being made good by the use of a thicker packing. This arrangement of packing a plunger makes a very compact design for a water-end, but the packing is troublesome to adjust or renew, because the back-head of the pump must be taken off and the pump drained before the stuffing-box can be reached.

To obviate this somewhat troublesome detail in manage-

ment, glands for internal stuffing-boxes have been furnished with a bale extending alongside, beyond the plunger and through the back-head, with suitable adjusting screws or other device on the outside of the pump. This greatly facilitates the adjustment of the packing, but does not make its renewal any the less difficult or disagreeable.

Outside-Packed Plunger-Pump with Central Diaphragm.—Another design for a plunger-pump is shown in Fig. 21. This is an adaptation of the water-end described

FIG. 21.

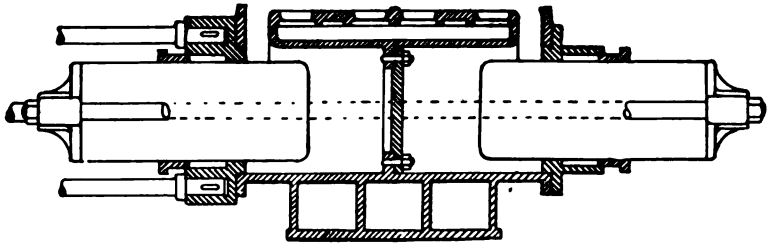


in the preceding section. The heads at both ends of the water-cylinder are provided with stuffing-boxes, as shown; the plunger-rod connects the two plungers so that their movements are coincident. The plunger-rod passes through a rigid bearing, which is bolted to an annular ring included in the water-end casting. This bearing may be bushed with brass, but generally this is not done. The leakage past the bearing is quite trivial, and is not seriously taken into account.

An outside-packed plunger-pump with tie-rods is shown in Fig. 22. It is a favorite one for tank-service or general supply in rolling-mills, steel-works, iron-furnaces, etc. The water-end has a central partition which divides the cylinder into two chambers, each having its own plunger and

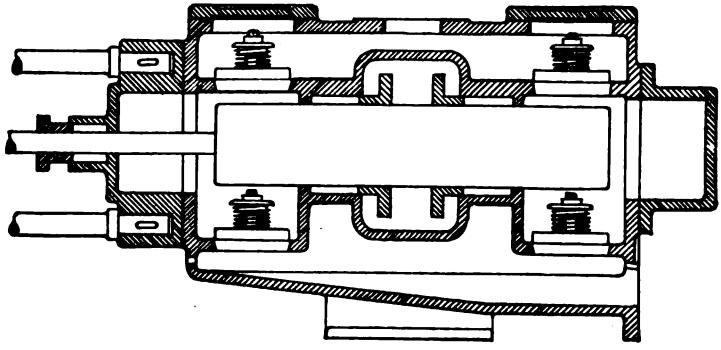
stuffing-box. The plungers are fitted with tie-rods, which serve to transmit the power from one plunger to the other, and to make their strokes coincident.

FIG. 22.



A centrally-packed plunger-pump is shown in Fig. 23. The principle of operation is the same as other plunger-pumps. One advantage which a pump of this kind has over the one described in the preceding section is that less room is required for the same displacement. The stuffing-boxes are not so accessible in this design as in the former one, but no

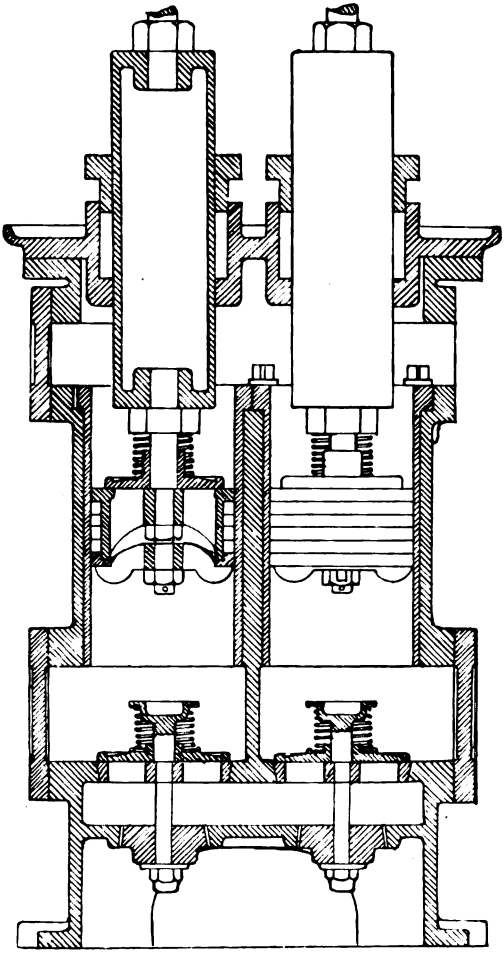
FIG. 23.



difficulty is experienced in adjusting or putting in new packing. An additional stuffing-box is required for the rod necessary to drive the plunger. This is an excellent design for a packed plunger water-end, and one which rarely fails to give complete satisfaction.

Bucket- and Plunger-Pump.—For any other purpose than simply that of lifting water the bucket-pump is not sat-

FIG. 24.



isfactory, because of its intermittent delivery. If, instead of discharging into the atmosphere immediately above the level of the bucket at the top of its stroke, as such pumps usually

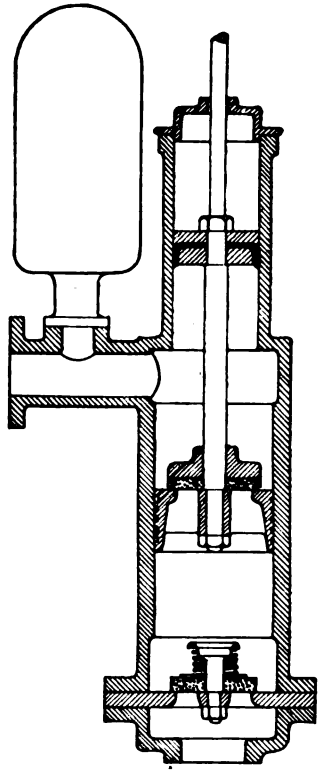
do, the delivery-chamber be lengthened sufficiently to fix a stuffing-box through which a plunger shall work, as illustrated in Fig. 24, it will be shown that a double delivery can be had with a single-acting pump. To begin with, the area of the plunger must be one-half that of the bucket, and the stroke of each must be the same, and coincident; the operation will then be as below:

We will assume the bucket to be at the bottom of its stroke and the pump fully charged with water; then by its upward movement a vacuum will be formed underneath the bucket, the water will flow into the pump-barrel from below and fill the empty space; let us assume that this volume be one gallon; on the return-stroke of the bucket all the water which flowed into the pump-barrel underneath the bucket (one gallon) passes through the bucket, past the valve, into the chamber above it; if this were an ordinary bucket-pump no overflow would occur on this its downward stroke, but the pump we are now considering is fitted with a plunger one-half the area of the bucket, and having a stroke coincident with that of the bucket, the plunger has by its downward movement displaced its volume of water (one-half gallon) in the upper chamber, this displaced water passing off through the delivery-pipe. The next stroke, upward, brings another volume of water into the lower chamber equal to one gallon, but at the same time that this water is being lifted the upper plunger is being withdrawn, and instead of one gallon of water passing through the delivery-pipe, only one-half gallon has been delivered at the completion of the upward stroke; now the downward movement of the bucket transfers the water (one gallon) from underneath it to the chamber above, the plunger descending at the same time with the bucket forces its volume (one-half gallon) out of the upper chamber; the effect of which is to convert a one-gallon, single-acting pump into a half-gallon, double-acting pump.

Bucket- and Piston-Pump.—A pump such as just described need not of necessity have a plunger; the same effect

can be had if there be two cylinders, one above the other, as shown in Fig. 25. These cylinders must have a ratio of area of two to one as in the former case, the larger or bottom

FIG. 25.

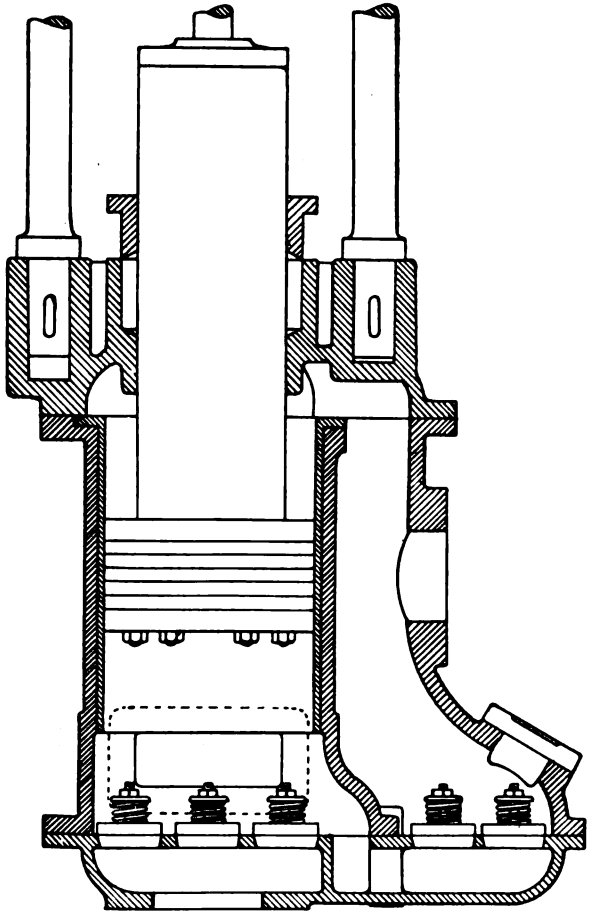


cylinder to be fitted with a bucket, and the smaller one with a piston. The operation of this pump is precisely similar to the bucket- and plunger-pump, already described.

Piston- and Plunger-Pump.—This is commonly known as the differential plunger-pump. It is, however, but a modification of the bucket- and plunger-pump, already described. The piston in this pump is not fitted with a valve, but is solid

and furnished with layers of packing or not, as the circumstances may require. This design differs from others of its class in the fact that a water-passage, or port, must be had

FIG. 26.



leading from the delivery valve-chamber to the end of the stroke of the piston in its barrel. Fig. 26 is a representation of such a pump arranged for use vertically. All the lifting of the

water is done at the upward movement of the piston; when the piston descends all the water below it is forced into the delivery-chamber; thus far it is but a single-acting pump. If we now consider the action of the plunger, it will be seen that during the movement of the piston downwards the plunger is entering the upper part of the pump-cylinder, through a stuffing-box, from the atmosphere; this plunger, being half the area of the piston, and of the same stroke, displaces only one-half the water forced into the delivery-chamber upon the next upward stroke of the piston; the other half of the water is discharged because the displacement of the piston is made complete by the withdrawal of the plunger, the delivery-valves being shut; these valves, as well as the suction-valves, open but once during a double stroke of the pump. It needs no further description to show that the lower part of this pump is single-acting, while the upper part containing the differential plunger is double-acting when operating in combination with its piston.

CHAPTER III.

PISTON- AND PLUNGER-RODS.

RODS which enter a water-cylinder through a stuffing-box must be round, straight, and well finished, and must also be free from seams and ridges, as both of these are likely to tear the packing.

For clear, cold, and fresh water, there is probably no better material for pump-rods than cold-rolled steel, and for this reason pumps are usually supplied with rods made of this material, unless for certain reasons some other material is to be preferred.

The tensile strength of cold-rolled steel piston-rods will average about 75,000 pounds per square inch of section. The elastic limit is increased somewhat by the process of finishing, and more nearly approaches the ultimate strength of the steel than is the case in ordinary bars. The superiority of cold-rolled rods consists in their almost absolute uniformity to size from end to end, much more so than can be had by ordinary turning and finishing. The hard, smooth surface produced by the finishing-rolls is the very best surface for the packing.

Piston- and plunger-rods are usually made much larger in diameter than simply to furnish the requisite strength for doing the work; if a rod breaks, the pump is useless until a new one can be supplied. This is an accident which now rarely occurs, so that it is not a common practice to keep spare rods on hand. Pump-rods must also be large enough to do the

work without vibration or tremor, as it would be next to impossible to keep a stuffing-box tight if it had a lateral motion in any portion of its stroke.

The size of a rod in a trade pump is usually selected for 150 pounds water-pressure upon the piston or plunger; in addition to this there is always an uncertain quantity to be added for water-hammer and other unusual strains. The diameters given in Table I. average equal to those in common use for the corresponding water-pistons or plungers at 150 pounds water-pressure and for pumps of 12 to 18 inches stroke.

TABLE I.

DIAMETERS OF COLD-ROLLED STEEL PISTON- AND PLUNGER-RODS OF 65,000 POUNDS TENSILE STRENGTH, FOR PUMPS OF 12 AND 18 INCHES STROKE.

Water-Piston or Plunger.			Cold-Rolled Steel Rods.			
Diameter. Inches.	Area. Square Inches.	Pressure at 150 Pounds.	Diameter. Inches.	Area. Square Inches.	Pressure per Square Inch on Rod. Pounds.	Strength in Pounds of Rod at 1-10.
4	12.57	1,885	1⅜	1.49	1265	9,865
5	19.64	2,946	1½	1.77	1664	11,505
6	28.27	4,241	1⅝	2.07	2063	13,455
7	38.49	5,774	1¾	2.41	2698	15,665
8	50 27	7,541	1⅞	2.76	2732	17,940
9	63.62	8,543	2	3.14	2721	20,410
10	78.54	11,781	2¼	3.98	2960	25,870
12	113.	16,950	2⅜	4.43	3826	28,759
14	154.	23,100	2½	4.91	4704	31,915

It is customary to assume 65,000 pounds as the average tensile strength of mild steel, and this figure is taken instead of the higher one given in the preceding paragraph relating to cold-rolled steel. A factor of safety of ten is allowed, as it is the one usually employed in all pump calculations.

TABLE II.

DIAMETERS TO WHICH COLD-ROLLED STEEL RODS MAY BE TURNED FOR BOTH PISTON AND PLUNGERS.

Diameters in Inches.				Area at Root of Thread. Square Inches.	Strength at Root of Thread in Pounds at 1-10.
Water-Piston or Plunger.	Pump-Rod.	Bore of Piston or Plunger.	Root of Thread.		
4	1⅜	1⅛	.94	.69	4,485
5	1½	1¼	1.06	.88	5,700
6	1⅝	1⅜	1.16	1.06	6,890
7	1¾	1½	1.28	1.29	8,385
8	1⅞	1½	1.28	1.29	8,385
9	2	1⅝	1.39	1.52	9,880
10	2¼	1¾	1.49	1.74	11,370
12	2⅜	1¾	1.49	1.74	11,370
14	2½	2	1.71	2.30	14,950

Tobin bronze, so named after its inventor, is a somewhat recent alloy, of which copper is the principal ingredient. This material has been used by the author in pump construction, for piston-rods and other details, and, so far as his experience goes, it fully bears out the claims of Mr. Tobin, as possessing great strength, toughness, and uniform texture. It can, when heated, be forged into any required shape as readily as steel. It works well in a lathe, and is susceptible of a high polish.

Tensile tests, by Professor R. H. Thurston, of cast specimens .798 inch diameter, 5 inches long, showed a tenacity per square inch of original section of 67,600 pounds; tenacity per square inch of fractured section, 73,160 pounds. The diameter of the fractured section = .767.

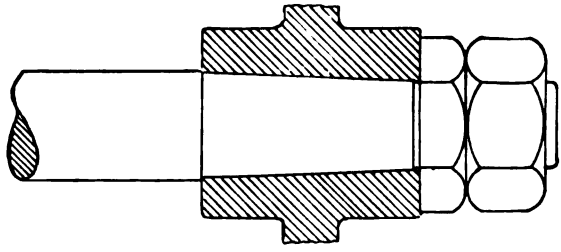
Samples of hot-rolled metal prepared from a ¾-inch cylindrical rod gave the following results when tested by Professor J. E. Denton under torsional strain: Tensile strength, 94,550 pounds, with a stretch of 36.44 per cent. in 4½ inches length of specimen, having a diameter of ⅝ inch.

The factor of safety given in the above tables, together with the net strength, has made it appear that the rods were figured on too liberal a scale; but we have not yet considered

the method of securing the piston or plunger to the rod; there will be in all cases a reduction in diameter, which will also reduce the margin of strength. Several kinds of fastenings are shown in connection with a water-plunger, but it is obvious that the same fastenings will be equally applicable for water-pistons.

Pump-Rod Details.—To those more accustomed to steam than to hydraulic work, their first suggestion would probably be to taper the pump-rod as in Fig. 27, and, indeed, many

FIG. 27.

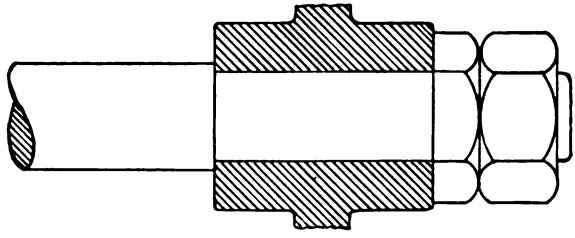


pump-rods are so fitted, but this method of fastening, so favorably known and practised in steam-work, has for one reason or another proven so troublesome in pump-work that it is now seldom employed. Any corrosion, dirt, or want of perfect alignment with the pump-rod will throw the piston or plunger out of line, and thus produce not only an uneven wear in a piston which is quite short for its diameter, but especially so in the case of plungers which have considerable more length than diameter; the friction first on one side of the plunger and then on the other side, as it passes the centre, makes a pump work badly, and is, moreover, a very difficult thing to correct. When a pump is new this difficulty is not usually experienced, especially in such shops as make it the practice to finish each plunger on its own rod; but it is afterwards, when repairs are needed, when the rods have become corroded, or when new pistons or plungers are required to fit an old rod,

or new rods to fit an old piston, that the trouble is usually had.

A common and a better method of securing a piston or a plunger to a rod is shown in Fig. 28, which simply con-

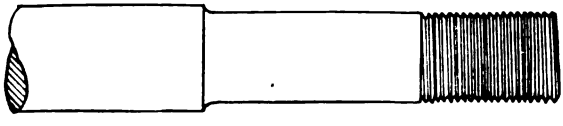
FIG. 28.



sists in turning down the rod to a diameter suited to the load on a pump. It does not matter whether the piston or plunger fits the rod or not; the rod being turned to a standard diameter to fit a nut, the hole in the piston or plunger being enough larger to slide on easily, a reamer $\frac{1}{64}$ of an inch larger will give an ample allowance.

A word of caution is needed in fitting rods with a shoulder: a sharp corner, as shown in Fig. 28, is wrong, and if the rod be so fitted it will be only a question of time, and usually not a long one, when the rod will break in the sharp corner; this

FIG. 29.



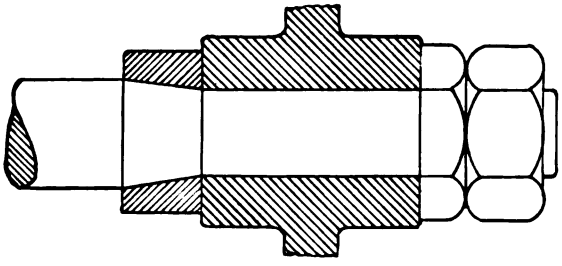
is especially true in the event that a steel rod be used. A fibrous iron rod will last longer than a steel one under the above conditions. There should be a liberal round or fillet in the corner, as shown in Fig. 29, and a corresponding curve in the piston or plunger.

Rods of the diameters given in Table I. may be turned

down to the diameters given in Table II. for their corresponding pistons or plungers.

If a larger area against the end of the piston or plunger be required, it can be had by turning a taper back of the plunger and fitting a collar, as shown in Fig. 30. The taper should be

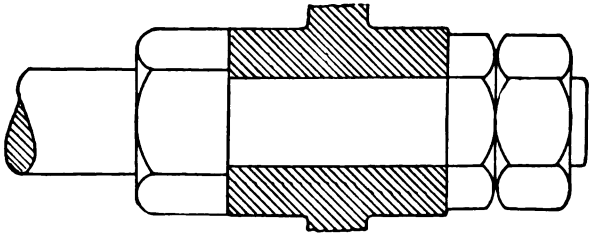
FIG. 30.



made on a lathe having a taper-attachment, and not by setting over the tail-centre. The collar should be reamed, and the rod carefully fitted to the collar, which latter should be faced off true in place to receive the end-thrust of the piston or plunger.

The same result may be secured by fitting the rod with the threads, as in Fig. 31. The larger thread must have its root

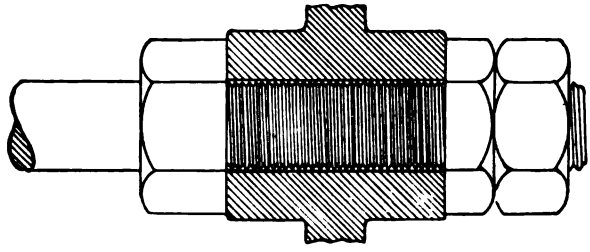
FIG. 31.



above the diameter intended for the plunger, and threaded for a hexagon nut, which should be faced off true on the rod-centres; the end of the rod is to be fitted with a thread below the diameter intended for the piston or plunger, as are all the others.

If the end of the rod be threaded, as in Fig. 32, no reduction in diameter will be required, and if thought advisable its diameter could be reduced at least one size from those given

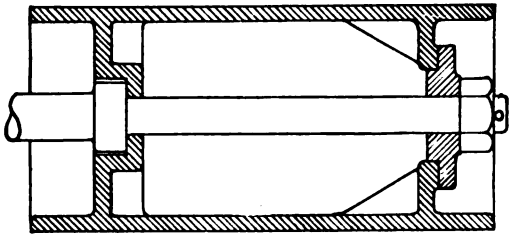
FIG. 32.



in Table II. In this arrangement the piston or plunger will be held by the faces of the nuts, which offer an ample bearing, all the strain coming on the thread.

In water-ends of large size, and others in which the rod is not continuous between the steam-piston and the water-plunger, the pump-rod may have a collar welded on, as shown in Fig. 33.

FIG. 33.

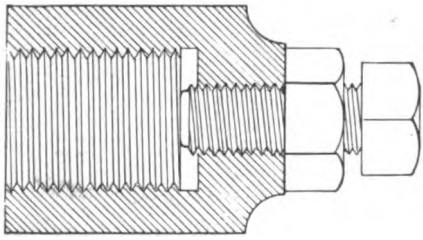


In such construction the rod is to be uncoupled at the cross-head between the steam- and water-cylinders; the plunger-rod cannot pass through the stuffing-box, therefore the plunger and rod are to be withdrawn entire through the rear end of the water-cylinder; provision must be made for such withdrawal when locating the pump in the building in which it is to be used. In regard to the size of collars, it is a common practice to make them once and a half the diameter of the rod, the length of the collar to be one-half its diameter.

Jamb-nuts are shown in the several illustrations for securing the nuts which fasten plungers to the rods. This is a simple and reliable device; a further precaution may be had by drilling a hole in the end of the rod and inserting a split pin to prevent the nut working off in case it should loosen.

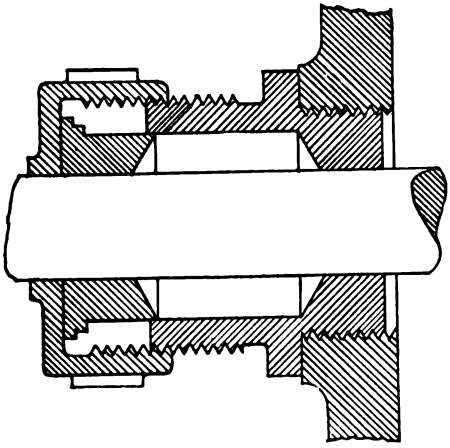
A special nut, as shown in Fig. 34, is sometimes used. It has an hexagonal body tapped to fit the end of the rod, and an extension piece smaller in diameter, drilled, tapped, and fitted with a set-screw for tightening against the end of the rod, this set-screw being fitted with a jamb-nut as shown. It possesses no advantage over the two nuts, and is much more expensive to make.

FIG. 34.



Stuffing-Boxes.—The design of stuffing-box shown in Fig. 35 is the one commonly used in small pumps; that is to say, for rods 1¼ inches in diameter, or less; but it is sometimes used for rods as large as 3 inches in diameter. These stuffing-boxes are usually made of brass, though some pump-makers make them of iron; they screw into a tapped hole in the cylinder-head or steam-chest. The cavity for the packing in stuffing-boxes of this design

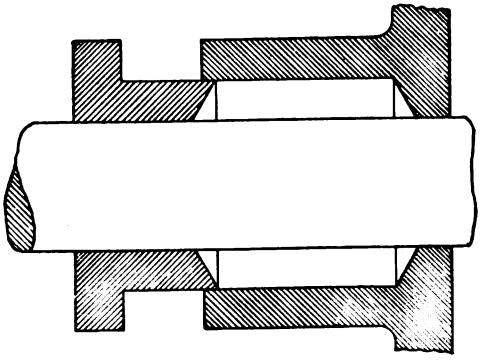
FIG. 35.



is not usually as liberal as in the design given in the next paragraph, therefore the packing will oftener require renewal.

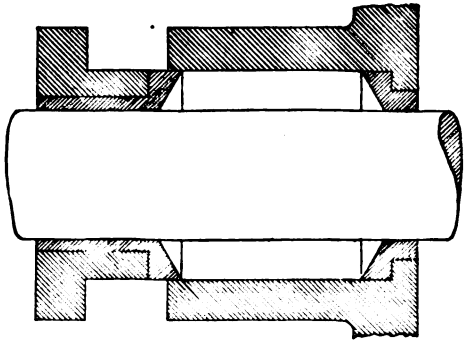
The commonest form of stuffing-box for sizes for 1½ inches in diameter and larger is shown in Fig. 36. The stuffing-box is cast together with the cylinder-head, and is bored concentric with the opening for the rod. The stuffing-box should always be deep enough to take in four strands of packing; the gland should be made to compress the packing, as it wears to nearly one-half its original bulk. The stuffing-box and gland here shown are intended to be of cast iron, which for ordinary service, and fresh water, will answer every purpose.

FIG. 36.



Stuffing-boxes and glands bushed with brass are shown in Fig. 37. For diameters of rod less than 2½ inches the gland should be made wholly of brass instead of inserting a bushing. The bottom of the stuffing-box shows the bushing extending through the thickness of the cylinder-head; this arrangement is a good one, but stuffing-boxes are frequently made with a brass washer, bevelled for the packing, but not extending through the head.

FIG. 37.



The question of bevelling the bottom of the stuffing-box and gland is one about which there is considerable difference of opinion; the writer favors the bevel for round packings, and thinks that even in the case of a machine-made square flax packing a tighter joint is had against the rod than is the case when the gland and bottom of the stuffing-box are flat; but for packings made up of layers of cotton cloth and india-rubber the bottoms of the stuffing-box and the gland should be flat.

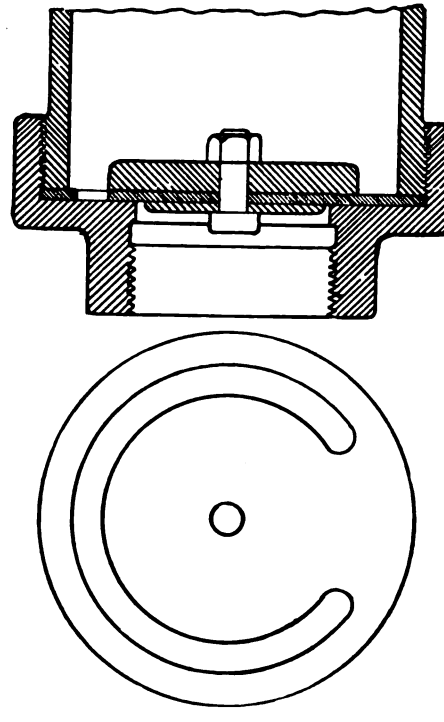
Metallic packings, so commonly used for steam-cylinder piston-rods, have not been found to be altogether satisfactory for pump-rods, and in consequence are but little used.

CHAPTER IV.

WATER-VALVES AND SEATS.

Clack-Valves.—The simplest form of valve for pumps of small size and low lift is shown in Fig. 38. It is largely used in hand-pumps and for small power-pumps for agricultural and house services. The lower illustration shows a plan of the valve. It is cut from a piece of leather about three-sixteenths of an inch thick, the outer diameter corresponding to that of the flange by which it is held, the inner diameter being enough larger than the opening into the pump to make a good joint under pressure; the clearance around the valve may be ⅛ to ¼ inch, depending on the size of the pump and the thickness of the leather. The allowance for hinging may be one-half the diameter of the valve. A metal plate under the valve nearly the size of the opening,

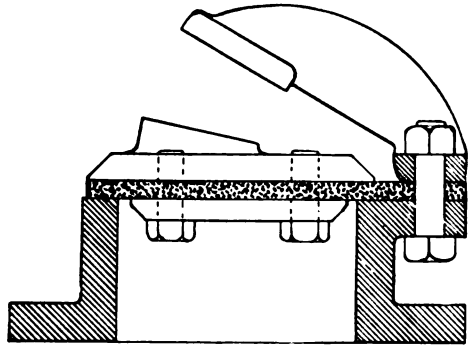
FIG. 38.



and another plate on the top as large as the diameter of the valve, should be securely fastened to each side of the leather by rivets or bolts. Nearly all English books on pumps mention the use of hippopotamus and rhinoceros hides for large valves; the same material is also referred to as being satisfactorily used for packing pump-pistons. Leather of this description is very sparingly used for pumps in this country, if at all.

A piece of india-rubber held at one edge, as in Fig. 39, is often used in large pumping- and sewerage-engines. The openings are usually rectangular, having a length several times that of the breadth; this is a simple and durable valve. The rubber may be 5/8 to 1 inch thick, depending on the service and the general dimensions of the pumps. The guard should have a generous curve to prevent breaking the rubber at the line of flexure.

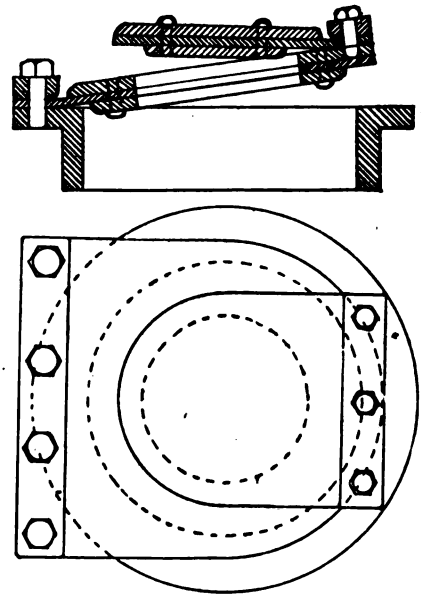
FIG. 39.



An interesting experience with clack-valves is furnished by Mr. Henry Teague (1887), in a paper read before the Institute of Mechanical Engineers, England. Mr. Teague had occasion to remove a 15-inch clack in 1863, substituting a new one with one-third of its area cut out in the centre of the flap; the aperture so made was covered with a supplementary leather clack hinged upon the main clack at the edge opposite the main hinge, as shown in Fig. 40. By this means the concussion, which was the cause of making the change, was so completely prevented that, by placing the hand or ear in contact with the clack-box, not the least tremor or sound was perceptible, and the time of closing

could not be detected. The leather of the small supplementary clack continued in constant action for seven years before requiring to be changed, and that of the main clack for thirteen years.

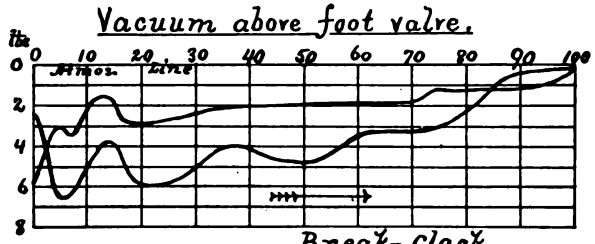
FIG. 40.



Another experience occurred in 1866 with a 14-inch pump, in which Mr. Teague substituted for double-beat valves others similar to the illustrations in the preceding paragraph, whereby the action of the pump was greatly improved, as shown by the two pairs of diagrams which are reproduced in Figs. 41 and 42, which were both taken under similar circumstances when the pump was travelling at 160 feet per minute. These diagrams were taken with an indicator having a spring of 10 pounds per inch of range. Fig. 41 shows the vacuum above the clacks, which, at the commencement of the stroke, is 5½ pounds per square inch for the break-clack, as shown by the lower line, and 7 pounds for the double-beat valve, as shown by the upper line. Fig. 42 shows the vacuum below the clacks, which began at 4 pounds per square inch in each case. The suction-pipe was 13 inches in diameter, and ran 40 feet horizontally from the pump before dipping into the tank, in which the water-level at the time of taking the diagrams was 6 feet below the bottom clack. In taking a number of diagrams similar to those shown in Figs. 41 and 42, it was found that scarcely any two were exactly alike. This was

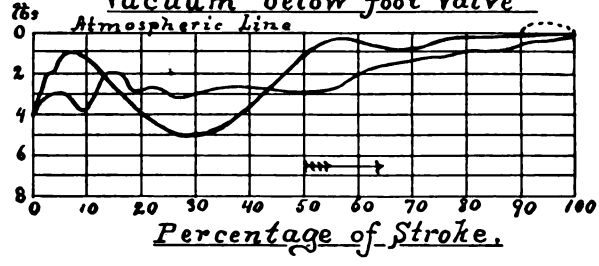
attributed, in most instances, to the undulatory motion of the water in the tank from which it was being pumped. The tank was 70 feet long by 40 feet wide, and the undulation was

FIG. 41.



Break-clack.

FIG. 42. Double-beat Valve.



sufficient at times to raise the indicated line above the atmospheric line at the finish, as shown by the dotted line in Fig. 42.

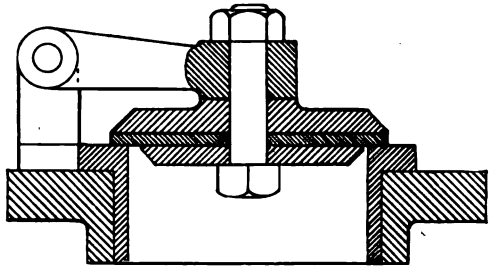
In some pumps with a vacuum of about 7 pounds per square inch, or 16 feet head of water at the commencement of the up-stroke, the indicated line has been raised several feet above the atmospheric line at the finish. To obviate this evil with a long horizontal suction-pipe, it occurred to him to interpose at the foot of the pump a vacuum-vessel for the long horizontal pipe to deliver into, the vessel containing sufficient vacant space to spare for taking up the fluctuations of the water, which are thereby prevented from affecting the working of the pump. This addition answered the purpose thoroughly, the pumps working with the greatest ease when so supplemented.

The break-clacks (Fig. 40) were found to work incessantly

for five years without changing a leather and without showing the least sign of leakage, under 350 feet head of water, or 150 pounds per square inch, and there is not the slightest concussion, the time of closing not being perceptible. For a velocity of 160 feet per minute of the pump Mr. Teague found the weight of the flap should be about 2 pounds per square inch. On the top of the second or supplementary clack a third of still smaller diameter can be added, and even a fourth or more hung alternately on opposite sides.

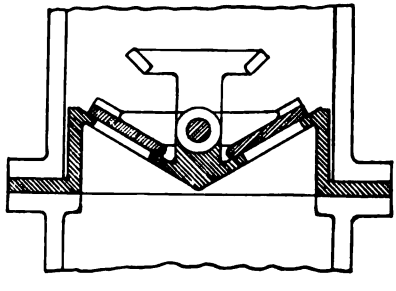
A hinged clack-valve made of metal with a leather face is shown in Fig. 43; this does not differ essentially in its action from the valve described in a preceding paragraph (Fig. 39), except that a fulcrum is usually above the valve, and should be as far removed from it as the design will permit; the greater the distance the better will be the flow through the valve-seat and the less will be the angle of lift.

FIG. 43.



A butterfly-valve is shown in Fig. 44. It is a modification of a pair of clack-valves hinged usually at a common centre. These may be of any convenient shape, and arranged in any manner best suited to the requirements of the pump. The valves are usually of metal, faced with leather or india-rubber; it is not

FIG. 44.



often that the valve and face close metal to metal,—probably more on account of the noise than anything else.

Clack-valves and butterfly-valves should be provided with stops to prevent their rising any higher than is actually necessary for the proper admission of the water into the pump; the angle of the valve should be about 30°, but must not in any case exceed 45°, above the seat.

Fulcrums for Clack-Valves.—The central pin, or fulcrum, should be of brass, and of liberal diameter to prevent

FIG. 45.

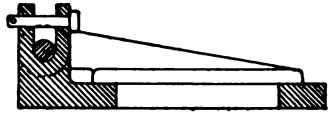
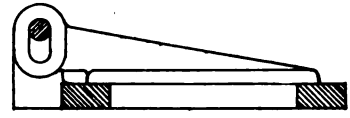


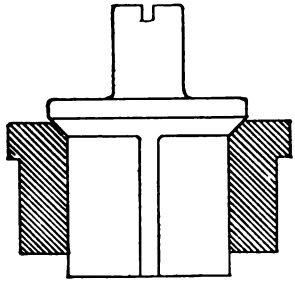
FIG. 46.



breakage. The wings, or valves, should have the holes oblong, at right angles to the face of the valve, so as to permit the valve to lift from ⅛ to ¼ inch vertically at the hinge before it begins to turn on its fulcrum. (Figs. 45 and 46.) This kind of valve is suited only to slow-moving pumps by reason of the long time it requires a valve to seat, and the loss of water occasioned by slip during the interval of seating.

A wing-valve, as shown in Fig. 47, is largely used in pumps for feeding boilers, and in hydraulic pumps for high pressures. The valve and seat should be made of hard gun-metal. The seat may be driven in straight, and is held by the friction of its sides in the hole in which it is driven; sometimes the seats are fitted with an external taper, but this is not the usual practice. The face of the valve and seat are tapered 45°, and are readily fitted and ground together. This valve may have

FIG. 47.



a spring fitted to its back to hasten the seating, but this is not always done.

Wing-valves sometimes have the lower portion of each wing on a curve, as shown somewhat exaggerated in Fig. 48, the object being to give the valve a partial rotation at each stroke of the pump, and thus compel the valve to seat itself in a new place at each stroke.

FIG. 48.

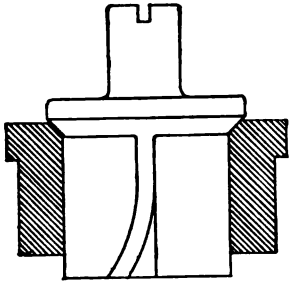
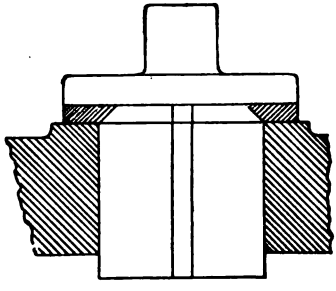


FIG. 49.

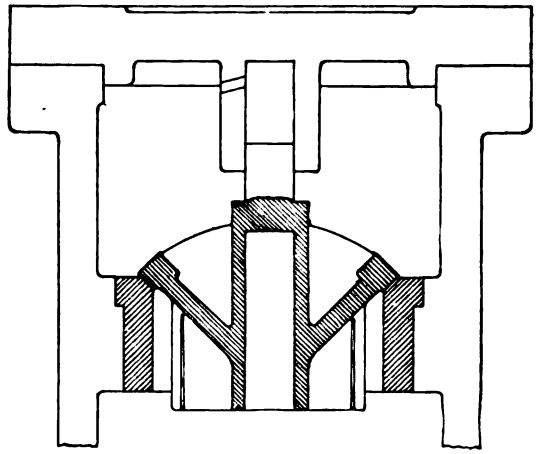


A wing-valve with a flat seat and a leather face, as shown in Fig. 49, is sometimes used in hydraulic-pressure pumps where the pressure does not exceed 400 pounds per square inch. It works fairly well, but possesses no marked advantages over similar valves fitted with a bevel and seating metal to metal.

A conical mitre-valve, as shown in Fig. 50, is recommended by Björling, but, so far as the writer knows, it has not been adopted by any pump-manufacturer in this country. It consists of a conical shell, the sides of the cone being at an angle of 45°; the top part is turned to fit the bevel in the top of the seat; the lower part of the cone is furnished with three or more wings or guides, according to the size of the valve; these guides should be put in at such an angle that the valve at every rise will partially rotate so as to wear the seat evenly and prevent any grooves being formed. Inside the cone is cast a short spindle, which beats against a stop provided in

the pump-valve box. This design of mitre-valve is considered by him the best, the flow of water being brought into

FIG. 50.



the proper direction by reason of the cone causing less obstruction to its passage and producing less contraction of the fluid vein than the one referred to in the preceding section.

A spindle-valve with a flat seat, such as shown in Fig. 51, is not often used in pumps. It is, however, occasionally met

FIG. 51.

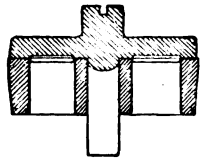
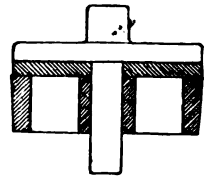


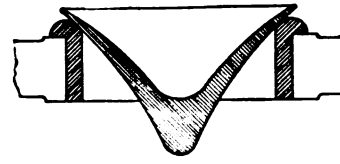
FIG. 52.



with in small pumps intended for light service, a disk of leather being sometimes attached to the valve to insure tightness when seated, as in Fig. 52.

A bell-valve for impure or thick fluids, such as tar, molasses, paper pulp, tan liquor, etc., is shown in Fig. 53. It has a perpendicular lift, but is not fitted with guides. It receives its name from a resemblance to an inverted bell. An inspection of the engraving will show its construction, and especially the concentration of weight below the valve-seat to insure its proper seating. The valve is suitable for slow-moving pumps only, and, so far as the writer is aware, they are not in use in this country.

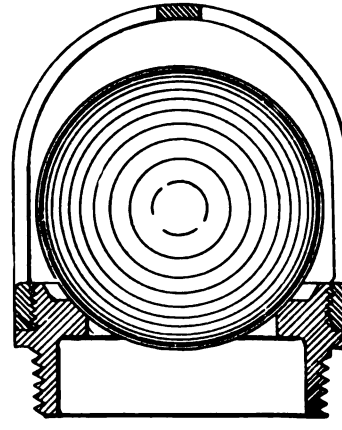
FIG. 53.



Ball-valves are used in considerable numbers, but the general impression is that they are not a regular, but a special, valve.

They are quite commonly in use in sugar-house pumps, filter-press pumps, and especially in artesian- or deep-well pumps. Fig. 14 shows the arrangement of a bucket of a deep-well pump, and Fig. 54 a design of a ball-valve and seat, for general service.

FIG. 54.



The ball and seat should be made of tough gun-metal, the guard should screw to the seat, and be so designed that it will have three or four ribs to guide the ball vertically as well as to limit the height of its lift, which should not exceed ¼ of an inch. The width of the guides should be as little as consistent with safety, so as not to obstruct the flow of the liquid above the valve-seat.

Ball-valves are sometimes made with an iron or lead core and covered with india-rubber

to a depth of ⅜ to ⅝ of an inch thick, depending on the size of the ball. This arrangement is not favored by the writer except for very light service. In his experience balls of this kind never gave satisfaction, but did give a great deal of annoyance at an average pressure of 100 pounds per square inch, the trouble being that the rubber covering would split and allow the metal core to pass through the seat or wedge fast in it under pressure. For cold water and light service, say 25 pounds pressure, the ball may be wholly of rubber, or weighted with a metal core to make it seat with greater promptness; but for higher pressures only gun-metal should be used. Ball-valves must be firm in order to retain their shape, but when made of gun-metal they need not exceed one-half of an inch in thickness for pressures up to 1000 pounds per square inch; care must be exercised in lightening balls that they are not made so thin as to prevent seating promptly.

Disk-valves made of india-rubber are in more general use than any other type of valve in this country. Valves for cold water should be of the best quality of india-rubber and vulcanized sufficiently to give the valve firmness, yet be sufficiently elastic to permit bending at right angles and regain its shape. For hot water a composition of india-rubber and graphite makes an excellent valve. A valve of this composition is hard and inelastic, and will not adapt itself to inequalities; it is necessary, therefore, to see that the valve-seat is true and flat. The valve should be scraped true to a surface-plate, and then the seat carefully fitted to the valve. These two valves are usually distinguished as a cold-water valve, meaning the india-rubber valve, and a hot-water valve, meaning the hard composition.

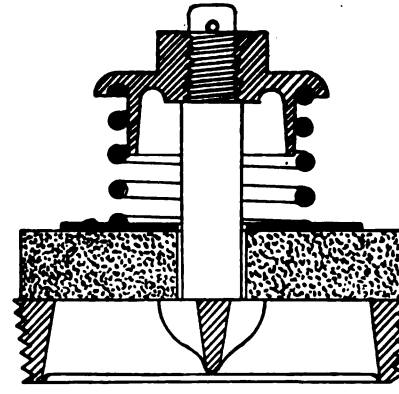
Disk-valves are usually made from 2 to 6 inches in diameter; sometimes larger diameters are employed in mine and other special pumps, but the general conclusion reached by builders and users alike is that rubber valves should not exceed 4½ inches in diameter. The four sizes more in use

than any other are 3, 3½, 4, and 4½ inches, and probably one-half of all the india-rubber valves in use of this type are 3 inches in diameter. The following proportions for india-rubber valves have given satisfactory results:

2	inches	diameter	×	⅜	inch	thick	×	½	inch	hole.
2½	"	"	×	7/16	"	"	×	½	"	"
3	"	"	×	½	"	"	×	9/16	"	"
3½	"	"	×	⅝	"	"	×	⅝	"	"
4	"	"	×	⅝	"	"	×	11/16	"	"
4½	"	"	×	¾	"	"	×	13/16	"	"
5	"	"	×	¾	"	"	×	13/16	"	"

Valve-Seat and Spindle combined.—The valve-seat shown in vertical section in Fig. 55 is one which has long been in use by the writer, and has given such excellent satisfaction that he gives it his fullest endorsement. The seat is threaded to screw into a corresponding taper-hole tapped in the valve-deck of the water-end of the pump; a taper of one inch to the foot will answer; the threads may be 8 to the inch for all sizes up to 4½ inches diameter. The valve-stem is included in the same casting with the seat; it is turned and polished so as to offer no resistance to the movement of the valve; an allowance of 1/16 of an inch is given for clearance between the valve and stem. The top of the stem is threaded with a taper-thread, the guard-nut being tapped with a corresponding taper-tap so as to screw down firmly, making a much tighter joint, and one less likely to jar or work loose than when parallel threads are used. As a precautionary measure a hole is drilled immediately above the guard-nut,

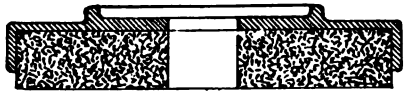
FIG. 55.



and a split pin inserted to prevent the nut working off and the valve getting adrift in the water-end. A brass plate $\frac{1}{32}$ of an inch thick for a 2-inch valve, and $\frac{1}{16}$ of an inch thick for a 4½-inch valve, is put on the back of each valve to distribute the pressure of the spring over a larger area, and to prevent the spring wearing a groove in the back of the valve. This plate need not be more than three-fourths of the diameter of the valve.

Valve-Cap.—A brass casting, as shown in Fig. 56, extending over the top and down the sides of a valve, has been used by the writer in a number of special cases with excellent results. There is no other objection to using it on all disk-valves of india-rubber except that of increased cost.

FIG. 56.



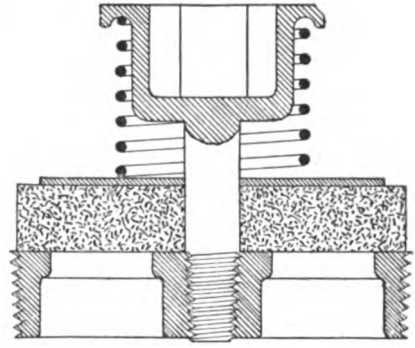
Springs.—No general directions can be given for the stiffness of springs, but if the 2- and 2½-inch valves be fitted with springs of No. 12 brass spring wire, the 3- and 3½-inch valves with No. 10 wire, and the 4- and 4½-inch valves with No. 8 wire, good results may be expected for the ordinary run of service. The diameter of the spring may be one-half that of the valve, and if made with five coils will have the proper elasticity.

A metal disk-valve, as shown in Fig. 57, will interchange on the same seat with the valve, spring, etc., just described. These metal valves are frequently used instead of the hard-rubber valves for hot water.

FIG. 57.

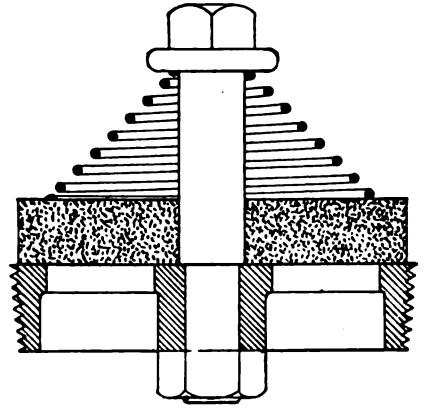
Valve-Seat with inserted Spindle or Guard.—A modified form of the design illustrated on page 57 is shown in Fig. 58. It does not differ materially from the seat and stem already described except that the seat is tapped and the guard screws into it, both threads being tapered as in the one previously described. This seat and guard is in quite general use, but is not so good a design as the former, because the guard is likely to, and often does, work loose, there being no practical method of preventing it.

FIG. 58.



Valve-Seat with threaded Spindle and Nut.—Another design for a valve-seat, guard, etc., is shown in Fig. 59. In this figure, it will be observed, the valve-seat is drilled with a straight hole, the guard has a shoulder, and is held down and in place by a nut underneath; this nut can be secured by a split pin if desired. The objection to this design is that the valve cannot be removed without unscrewing the nut, and in almost every pump the under side of the valve-seats are inaccessible in the suction-chamber, and it frequently happens that those in the force- or delivery-chamber are equally so.

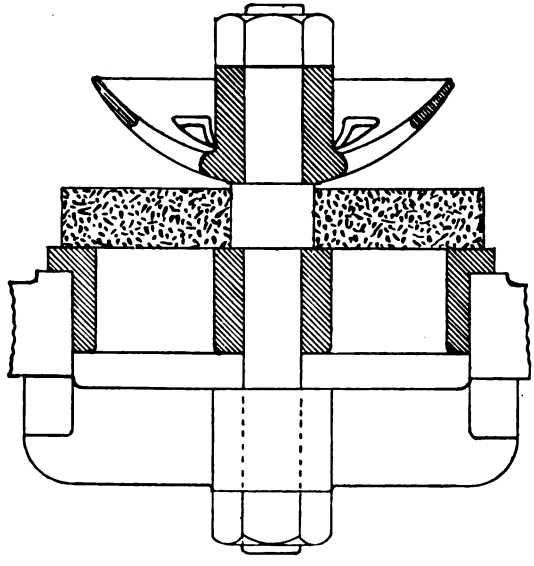
FIG. 59.



Conical Springs.—Attention is directed to the conical spring used in this illustration; this is a style of spring formerly in almost universal use, but it has been superseded within the past five or six years by the parallel spring, which has already been described. Some makers of pumps continue to use it, but the spring is defective in the fact that all the strain of the spring centres in the two upper coils, and breakages constantly occur at that point. In a parallel spring, as shown in Fig. 58, the tension is throughout its whole length, and breakages seldom occur.

Disk-Valve with Hemispherical Guard.—An india-rubber disk-valve with hemispherical guard, as shown in Fig. 60, is largely employed in pumping machinery in Europe, and

FIG. 60.

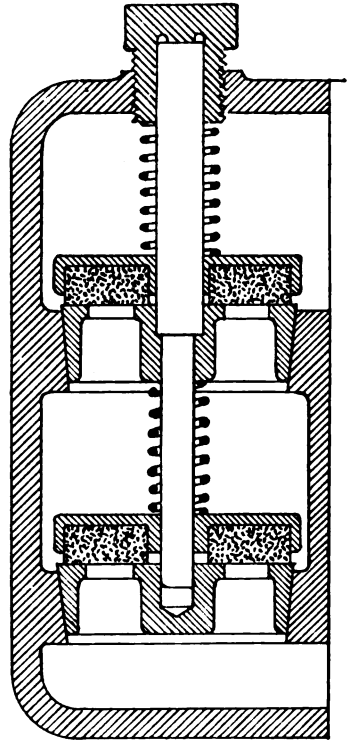


is often of large size, having long been in favor with pump-makers and marine-engine-builders. It is seldom met with in this country except in mining-pumps. It possesses no advan-

tage over the flat disk-valve with spiral spring, as shown in Fig. 55.

Securing Valve-Seats in Place.—A design of pump-valve shown in Fig. 61 is one which has long been in use by the makers of the Cameron pump. It consists of a brass shell filled with vulcanized india-rubber. The rubber forms the valve-face, and the metallic casing resists the pressure and protects the rubber from injury. It will be observed that the metal casing extends downwards and surrounds the stem. The valve-seats are driven in on a taper, one exactly above the other. One valve-stem is common to both valves. By unscrewing the plug on the top of the delivery-chamber the valve-stem can be withdrawn and the valves and springs removed for examination or repairs through the hand-hole plates.

FIG. 61.

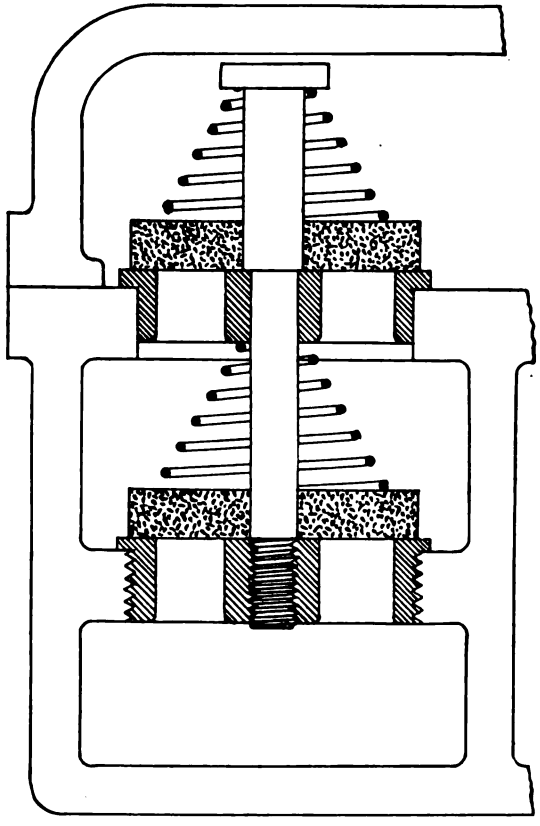


Another method of securing valve-seats and valves is shown in Fig. 62, which represents the practice of the makers of the Davidson pump. The suction-valve seat is screwed into the pump-casting, and is tapped to receive a spindle or valve-guard; the delivery-valve seat is centred by the upper hole, which is bored in line with the lower one. A shoulder is provided on the upper valve-seat. The spindle, or valve-guard, passes through the upper valve-seat, and is made to screw into the lower seat; a shoulder is provided on the valve-spindle for holding the upper valve-seat down upon its bearing when

the spindle is screwed sufficiently into the lower valve-seat. The examination or removal of the valves is easily had by the withdrawal of the valve-spindle.

Large valve-seats are often made with projecting wings or flanges for bolting the seat in place, somewhat after the style

FIG. 62.



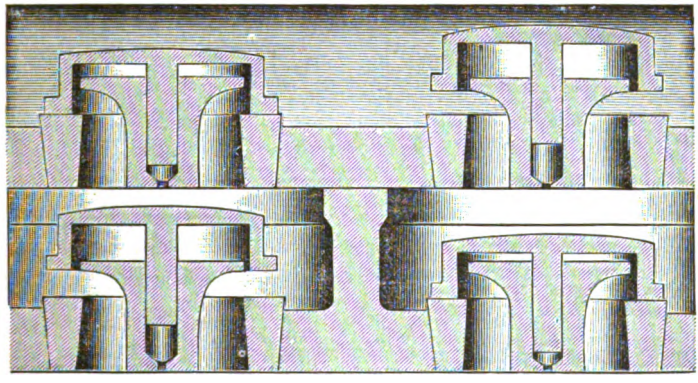
of Fig. 71. Standing-bolts are recommended rather than tap-bolts. Muntz-metal makes a good bolt; the nuts may be of cast brass.

For mine-pumps the valve-seats are often flanged and

inserted between the pump-flanges, as shown in Fig. 44, as well as in several illustrations in the chapter on mine-pumps. In designing a pump for high pressures with such a valve-seat, the lower pump-flange should have a diameter of opening the same as that of the valve-seat above it; this will give the latter a better support than if held by the flange alone.

An easy-seating water-valve is illustrated in Fig. 62, A. It consists of a cup-shaped valve with central guiding-pin, and a valve-seat and disk cast solid, as shown in cut.

FIG. 62, A.

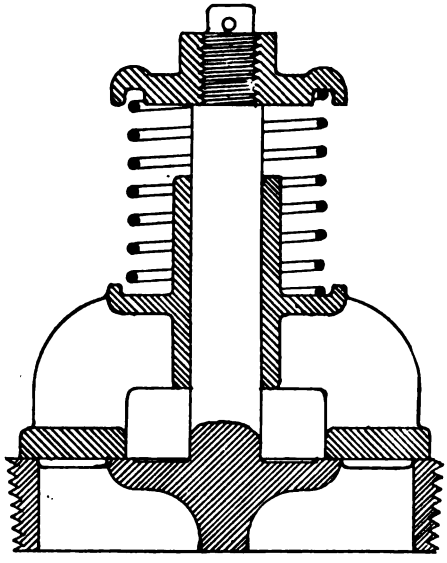


This disk is slightly larger than the aperture below it in the valve-seat, and causes the fluids pumped to be deflected at right angles, just the same as an ordinary water-valve does with its lift limited by a stop. The fluids, therefore, have power to raise the valve as high as the disk only, and for that reason a stop to limit the lift of the water-valve is not necessary. This water-valve does not strike against a stop in its upward movement. When it closes it does not seat with a harsh, abrasive action, but, being partially cushioned on the water between the disk and valve, it seats softly, and there is reason to believe it a durable and satisfactory water-valve.

The water-ways are large, and the valve has capacity equal to any other. This valve is secured by a recent patent, and is used in the Marsh steam-pump.

An annular valve is shown in Fig. 63. The seat has grids to give support to the central spindle, or guard, but which do not extend up to the valve-face. The valve is an annular ring fitted with wings which attach to a bored guide sliding on the valve-spindle. A spring, with nut, etc., is furnished as shown.

FIG. 63.



In this valve-seat the area is somewhat reduced under that of an ordinary valve of the same diameter, but having two circumferences for the discharge, the quantity of water which may be delivered is very much increased. Assuming the outer diameter of the opening through the valve-seat to be four inches and the inner one to be two and a half inches, there is then:

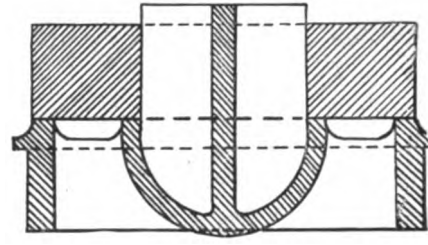
4 inches diameter = 12.57 inches area and 12 56 inches circumference.
2½ " " = 4.91 " " " 7.85 " "
Net area = 7.66 inches, and 20.41 inches total circumference.

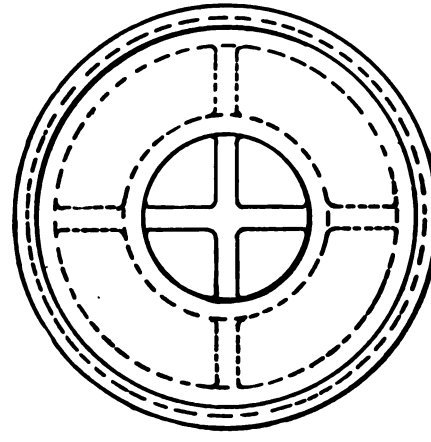
The lift required would be $7.66 \div 20.41 = .375$ of an inch to give the full area.

The annular valve shown in sectional elevation and plan in Fig. 64 is exceedingly simple and has much to commend it for

valves of medium size, and for pumps running at a moderate rate of speed. Weighted valves do not seat as promptly as those in which the initial impulse is given by a spring. The valve in this design is simply a brass ring guided by four wings, which are included in the same casting with the seat. The lift of the valve not being excessive, there will be little or no tendency to stick or to prevent prompt seating.

FIG. 64.



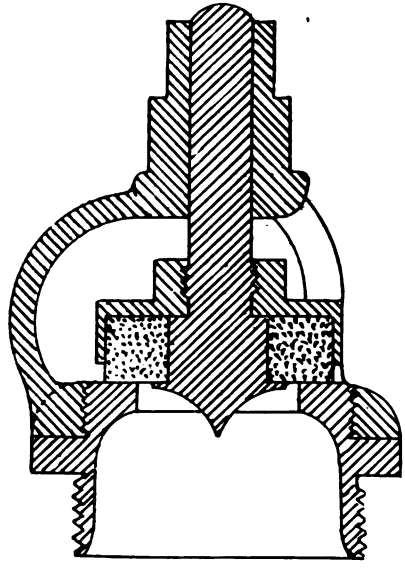


The Troy Valve.—Designers of pumping machinery have always endeavored to keep down the lift of a valve, and to get rid of the grids in the valve-seat; some of these efforts have been successful, others less so. Fig. 65 is an illustration of a valve and seat by the Holly Manufacturing Company; it has a 1¼-inch clear way of opening through the valve-seat, a flat rubber valve enclosed in a metal shell, which is top-guided; it is not furnished with a spring to assist it in seating. This may be said to fairly represent the practical limit to which the reduction in diameter of valves for large pumping engines has been carried out.

Valve-Guides.—Guiding a valve by a spindle issuing from the top of the valve and extending into a drilled hole in the cover of the valve-chamber, and at the same time having

wings to the valve fitting in a bored valve-seat below, as shown in Fig. 66, is occasionally met with, and is a practice which the writer does not consider a good one. The fitting of a valve and seat together is in itself an operation of the greatest nicety, and requires the utmost precision and freedom of movement to make its operation satisfactory. No such precision is had in fitting valve-covers, so there can be no certainty that the centres of the valve-seat and the drilled hole in the cover are in line with each other; if not, the operation of the valve will be faulty, and in consequence the action of the pump uncertain. Valves of this type should never be guided in two places, and as a matter of choice between the two guides, as shown in the engraving, the wings fitting the seat is one much to be preferred to that of having the spindle guided in the cover.

FIG. 65.



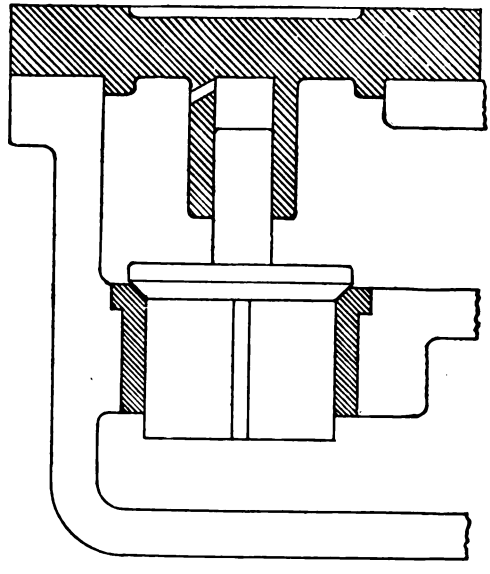
A top-guided valve without the lower wings is shown in Fig. 67; the top-guide is, as in the former case, the cover of the valve-box. This design is objectionable also, and for the same reasons as given in the preceding paragraph, there being no certainty that the guide for the valve-stem will be at right angles to the valve-seat, or that it will be exactly in line with the valve movement.

In all cases where it is thought to be necessary to have a mitred valve guided from the top, the valve-seat and guide should be self-contained, so that the seat, valve, and guide may all be fitted together before its insertion in place in the pump.

For valves having a leather or rubber face coming down upon a flat valve-seat the conditions are more favorable to satisfactory working, but on general principles the writer advocates having valve-seats and guides self-contained.

A built-up wing-valve of large diameter (360 mm.) is shown in Fig. 68, in section, with a partial plan showing the arrangement of the wings or guides. The valve-stem has a welded collar against which the flat disk for taking the pressure is fitted; underneath this disk is another one of rubber for making the joint on the valve-seat; still another metal disk is added, after which the wings or guides are placed on the same spindle, the whole being held in place by a through-going key, as shown. The details of this valve are good, the only objection being the unusual number of parts which compose it. An examination of the engraving will make clear that no derangement is likely to occur which would interfere with its proper working, except that of the backing out of the key; a split pin would make secure against such a contingency.

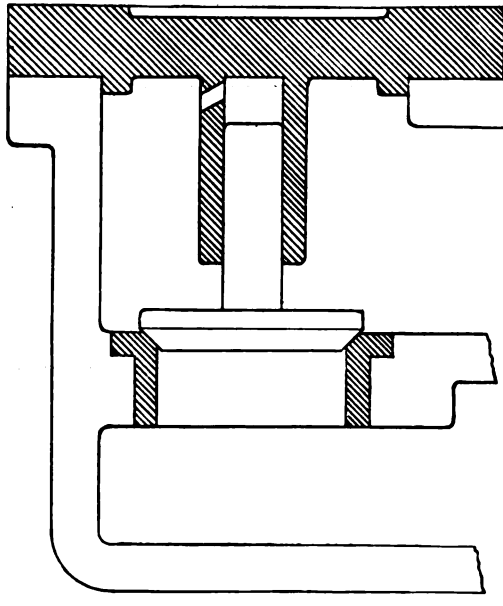
FIG. 66.



A rectangular valve with double seating is shown in Fig. 69. This is a form of valve not much used in this coun-

try; it is open to the objection that the seating of the valve on the grid always occurs at the same place, causing a rapid wear of the valve under what are usually thought to be very moderate pressures. The writer does not recommend this design except for light service, say for pressures not exceeding 25 pounds per square inch, although such valves are in use at pressures many times greater than the limit placed upon them above; nevertheless, they have not shown themselves durable under high pressure, and are quite inferior in many respects to circular valves, which are free to rotate about the central spindle or guide. Rectangular valves should always be of the very best quality of india-rubber to insure satisfaction, as they are subject to shearing action over the openings in the valve-seat, as well as the bending movement along the central line of fastening.

FIG. 67.

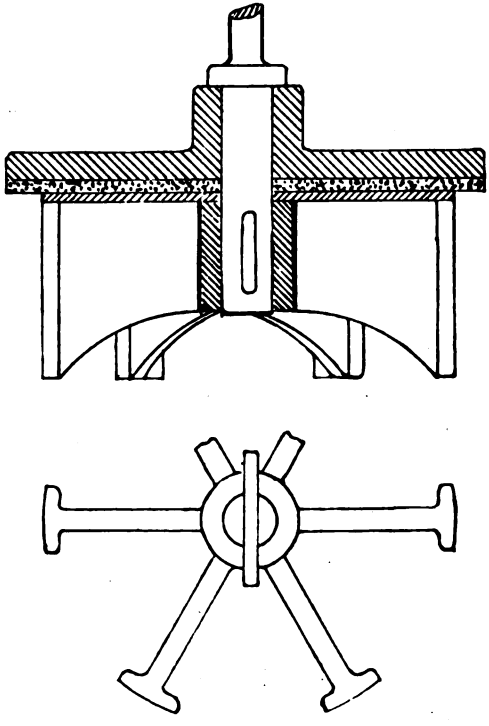


A multiple beat-valve with four beats or valves is shown in Fig. 70. This form of valve is not anywhere in very general use; in England, probably more than elsewhere; but, so far as the writer is aware, it is not used in this country in any prominent pumping-station. The main valve-seat may be secured to the valve-chamber casting by any one of several

methods; the central spindle has a length suited to the combined thickness of the four valves, together with a proper allowance for the lift of each.

The bottom valve has two faces, the lower one fitted to the valve-seat and the upper one fitted to receive the second valve, which is similarly fitted for the third valve, and thus for the fourth valve, which completes the series. There are four openings for the passage of water, so that after making allowance for the difference in the circumferences of the several openings, there will then be required a lift but little more than one-fourth over that, if one valve had to do the work.

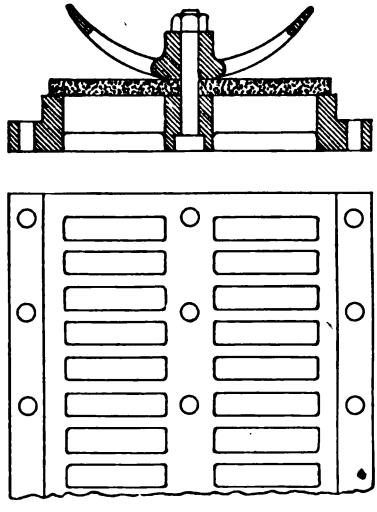
FIG. 68.



There does not seem to be an agreement of opinion among English engineers regarding this valve, some thinking highly of it, while others have taken them out and substituted valves of a different design. The principal objection seems to be that by reason of the large diameter as compared to the height the rings are liable to stick; the valve is also reported as being noisy in operation, which would seem to indicate intermittent or irregular seating, with the attendant jar and hammer in the water end of the pump.

A double-beat india-rubber valve is shown in Fig. 71. The main valve-seat is bolted over a suitable opening in the valve-chamber; this valve-seat casting includes the grid for the lower annular ring, of india-rubber, loosely fitting around a central cylinder also included in the same casting, and forming the water-way to the upper valve. A curved guard controls the rise of the lower valve, the latter having a vertical lift of about ⅜ inch before coming in contact with the guard. A brass grid forms the upper valve-seat, and by a slight projection downward also serves to keep the large guard in place. A collared bolt and nut secure this upper grid in place, and forms the central spindle for the upper valve; the curved guard for this valve permits the latter to rise about ¼ inch before contact, the guard being secured to the vertical spindle as shown. This is a better form than a single large valve of the same diameter as the lower ring, because the circumferences of the two valves are much greater than the one large valve at the bottom, thus permitting the passage of the same volume of water with a greatly reduced lift, and in consequence smoother working than if all the water were required to pass the circumference of the larger valve only.

FIG. 69.

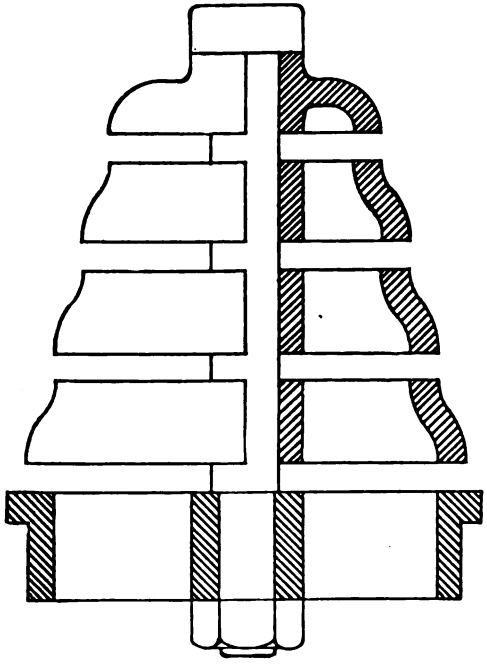


Cornish Double-Beat Valves.—These are largely used in England, and to a considerable extent in this country, but as for ourselves the general tendency is towards employing a number of small valves rather than one large one in water-

ends, so that, for the present at least, they may be regarded as out of style.

Through the courtesy of Mr. A. F. Nagle I am able to present a copious extract from his paper on "Cornish or

FIG. 70.



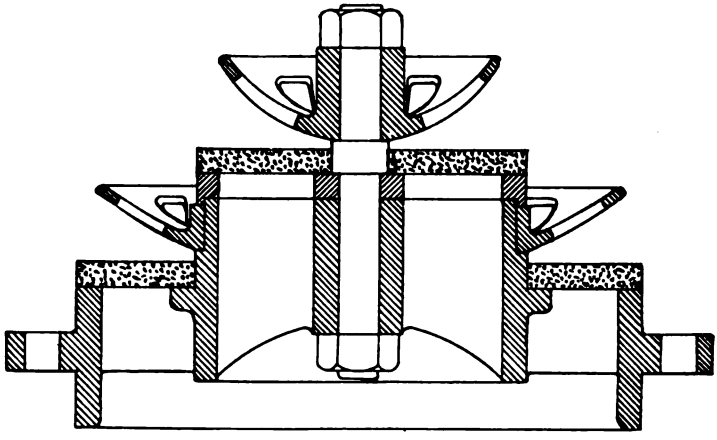
Double-Beat Valves," contributed to the American Society of Mechanical Engineers, and published in Vol. X. of its Transactions.

"The principal features to be considered are,—

1. The width of seat.
2. The unbalanced area.
3. Its weight.
4. Its lift.
5. Its form of body.

" 1. *The Width of Seat.*—What should be the width of a valve-seat? Theoretically, a knife-edge, so that the same area

FIG. 71.



should be presented to the water-pressure before as after it is lifted. Practically, only sufficiently wide to sustain the pressure

FIG. 72.

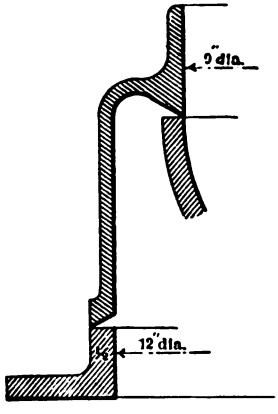
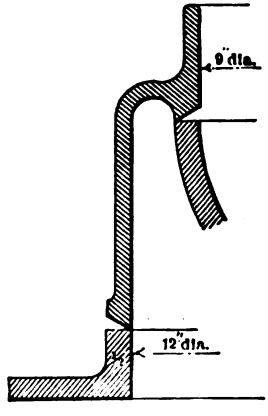


FIG. 73.



brought to bear upon it without injury to the metal. Brass should sustain a pressure of at least 1000 pounds per square inch of surface with safety and permanency. This is less

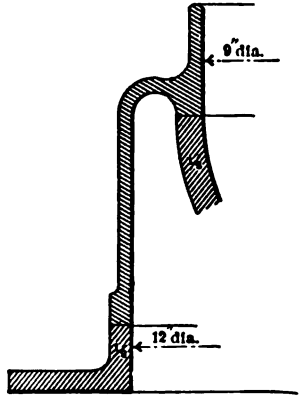
than one-thirtieth of its crushing strength, and only about two-thirds the pressure brought upon crank-pin journals.

"2. *The Unbalanced Area.*—With a wide seat it is impossible to know exactly what the unbalanced area of a valve really is. It may be that of either extreme between the inside or outside diameters, as indicated in Figs. 72 and 73, or it may be a yet worse case if the bearing should be perfect over its entire surface, like Fig. 74, where it may approximate to a vacuum between the faces.

"Even if we do not assume a vacuum to exist between the faces, it is still certain that something less than the normal pressure must be between the faces, or the valve would be in a leaky condition; and hence there must inevitably be required a greater pressure per square inch to start the valve than exists outside of it, and this condition is one which accounts for the shocks and noise of these valves.

"3. *The Weight of the Valve.*—I thought it was the weight of the valve, if free to move, which determined the velocity of discharge through it. If the valve is large in diameter compared with its lift, so that the velocity of approach becomes so small that it could be ignored, and its form of such gentle curves that no violent impingement occurs, then it would seem that the weight per square inch of unbalanced area must govern the flow or velocity; for it is this weight which is the equivalent of a pressure upon the water within the valve which causes the outward flow. And if this theory were correct, then the flow through it would have the same velocity at any position it might be in, and the valve should rise and fall in exact proportion to the changing velocity of the plunger. If, on the other hand, the valve be of irregular form, very light, and the velocity of

FIG. 74.



approach very great, then the calculation for its action would be very complicated.

"4. *Its Lift.*—This is practically answered in the last section. The velocity being determined by the weight, and always the same for the same weight, then its lift would naturally adjust itself to the changing speed of the plunger, so that the requisite water might be supplied to it or discharged from it. If, for example, a valve weighed one pound per square inch of its inside unbalanced area, it was reasoned that the velocity through the valve would be that due to this pressure, or, applying the well-known formula:

$$v = 8.03 \times \sqrt{h}.$$
$$v = 12.20 \text{ feet per second.}$$

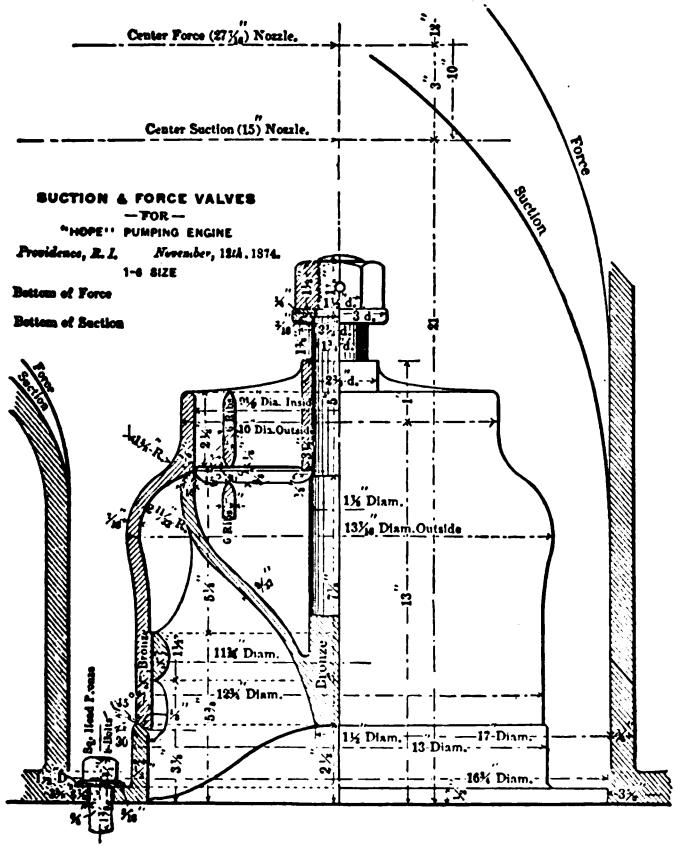
"The size of the plunger and its velocity, and the number of valves, now determine the lift of each valve.

"5. *The Form of the Valve.*—First of all, there should be no air-pockets such as are possible in such forms as are shown in Figs. 72, 73, or 74. The curves should all be of easy lines in order to avoid impact, it being reasoned that flat surfaces, particularly at the upper bend of the valve, would cause an impact which would make the valve rise more than that due to the pressure produced by the velocity.

"With these theories in mind I constructed the pump-valves for the High Service Pumping Engine at Providence, R. I. The engine is of the vertical compound type, with cranks exactly opposite each other (the first instance of the kind in this country, I believe, 1874) and geared 1 to 5, driving two horizontal double-acting plunger-pumps. For full description of this engine, see *Franklin Institute Journal* for September, 1876. The plungers are 17 inches in diameter and 4 feet stroke, and the greatest speed about 20 revolutions per minute. All the valves were of the same size, and only one for each inlet or outlet, and that was 12 inches in diameter at the lower seat, and 9⅛ inches at the upper. The seats were designed to be three-eighths of an inch wide, but the *seats*,

not the *valves*, were actually chamfered so that only one-eighth of an inch bearing-surface remained.

FIG 75.



"Fig. 75 is a vertical section of the valve, and Fig. 76 a full-size section of the seat.

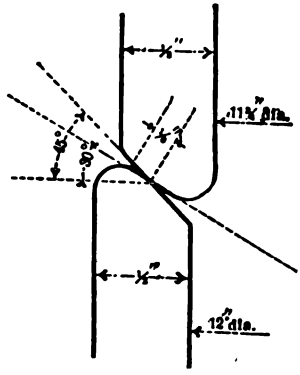
"The weight was 53.44 pounds in water, one-seventh less than in the air.

Mean net water-pressure	=	52 pounds.
Lower seat outside diameter 12¾″	=	127.68 square inches.
" " inside " 12½″	=	122.72 "

Upper seat outside diameter	$9\frac{1}{8}''$	=	65.40 square inches.
" " inside "	$9\frac{3}{8}''$	=	69.03 "
Net outside unbalanced area		=	62.28 "
" inside " "		=	53.69 "
Seat area		=	8.59 "

"The seat area is only 16 per cent. of inside area, and only $52 \times 62.28 \div 8.59 = 377$ pounds pressure per square inch of surface.

FIG. 76.



"Upon the theory of a perfect seating, the pressure required to open the valve would be $62.28 \times (52 + 15) \div 53.69 = 77.31$ pounds.

"I confess that I am somewhat sceptical as to the possibility of such perfect seating of a ground valve as to produce the condition of a vacuum, but somewhere between a vacuum and the water-pressure it must be, and I have assumed this extreme condition the better to illustrate a theory.

"These valves worked noiselessly at the greatest speed, and after six months' run the grinding-marks were not worn away.

"It will be observed that the valve weighed just about one pound per square inch of inside unbalanced area, and hence, if the theory advanced in section 3 were correct, the velocity of the water through it should have been 12.20 feet per second. It proved to be fully 20 feet per second.

"In order to learn how much truth there was in these theories, I took several indicator-cards directly from the valve itself.

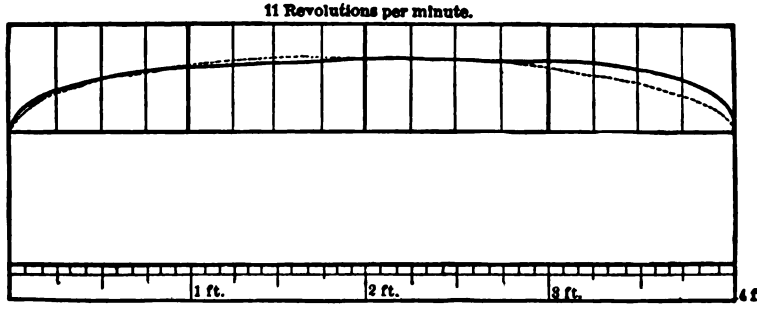
"These cards are reproduced at full size in Figs. 77, 78, and 79, and the dotted lines are added to represent the line of velocities of the plunger at all points of the stroke.

"The greatest lift attained at

11	revolutions per minute	was	$\frac{1}{8}$	inch	= .032	foot.
13	" "		$\frac{5}{32}$	"	= .039	"
18	" "		$\frac{9}{16}$	"	= .047	"

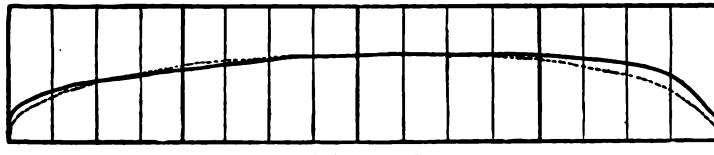
"It is not possible to know the exact diameters at which the discharge may be considered to take place, but I as-

FIG. 77.



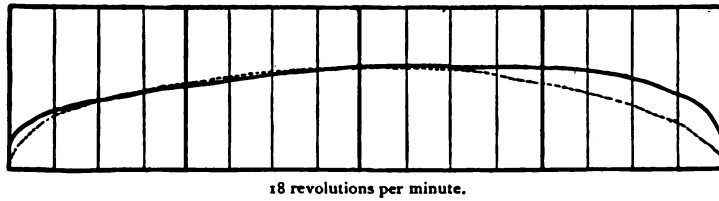
Position of piston.
Heavy line traced by indicator-piston attached to valve.
Dotted lines velocity of pump-plunger.

FIG. 78.



13 revolutions per minute

FIG. 79.



18 revolutions per minute.

sumed it for the lower seat at 12 inches, and the upper at 9⅛ inches.

"Circumference of discharge at

12 inches diameter	= 37.70 inches.		
9⅛ " "	= 28 27	"	
Total	= 65.97	"	= 5.50 feet.

"Area of discharge of valve at

11 revolutions	= 5.50 × .032 =	.1760	square feet.
13 "	= 5.50 × .039 =	.2145	"
18 "	= 5.50 × .047 =	.2585	"
Area of 17 inches plunger		1.576	"

"Maximum velocity of plunger at

11 revolutions	= 4 × 2 × 11 × 1.57 ÷ 60 =	2.30 feet per second.
13 "	= 4 × 2 × 13 × 1.57 ÷ 60 =	2.72 " "
18 "	= 4 × 2 × 18 × 1.57 ÷ 60 =	3.77 " "

"Displacement of plunger at

11 revolutions	= 1.576 × 2.30 =	3.6248 cubic feet per second.
13 "	= 1.576 × 2.72 =	4.2867 " "
18 "	= 1.576 × 3 77 =	5 9415 " "

"Velocity through valve at

11 revolutions	= 3.6248 ÷ .1760 =	20.60 feet per second.
13 "	= 4.2867 ÷ .2175 =	20.00 " "
18 "	= 5.9415 ÷ .2585 =	23.00 " "
It was calculated to be		12.20 " "
Head due to velocity of	20.60 feet =	6.60 feet, or 2.87 pounds.
" " "	20.00 " =	6.20 " 2.70 "
" " "	23.00 " =	8.20 " 3.57 "

Weight of valve per square inch of unbalanced area, 1 pound.
Ratio of weight of valve to pressure due to flow through the valve, about 1 to 3.

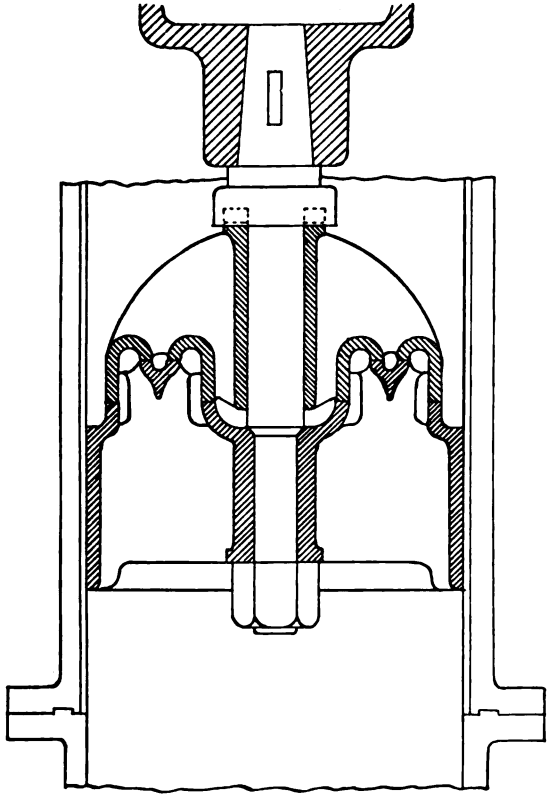
"The diagrams, as well as experience, showed:

"First, that the width of a valve-seat could safely be brought to a very narrow surface, probably much less than I made it ($\frac{1}{8}''$); for the pressure in this case was but 377 pounds per square inch of surface.

"Secondly, that the lift of a valve is exactly proportioned to the velocity of the plunger, if it is not too light so as to be brought to its stop before the maximum velocity of plunger is attained. The deviation from this theoretical curve, as shown in the cards, is attributable to the friction of the stem running to the indicator, and possibly somewhat to seat area, small as it is.

"Thirdly, that in the form of valve shown, the theory that the velocity of the water through the valve is that due to the head corresponding to the weight of the valve per square inch of unbalanced area, did not prove to be very near the truth. I can conceive of but one reason for this great variation from

FIG. 80.



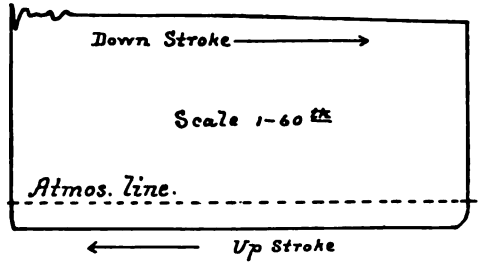
the theory assumed, and that is, the effect of the horizontal issuing stream diminished the vertical pressure. I think it is not improbable that there is a mathematical demonstration for the resultant vertical force due to an issuing horizontal stream from a curved aperture, and express correctly the relation of

lift to weight, but I have not attempted to thus solve that problem. I do not think that the friction of stem or force of spring is sufficient to account for the deviation.

"The valves were symmetrical and round in form, and were afterwards turned down and reduced in weight to 35 pounds in water, or .66 pound per square inch of inside unbalanced area, but I regret that I took no further diagrams. The narrow seats, and the quiet action, and the synchronous motion with the plunger, were the more important features in my mind at that time, and the question of weight of valve was left to experiment after all."

The four-seated valve shown in Fig. 80 is reproduced from an engraving in *Engineering* of a pump designed by Edward Easton & Co., London. It represents the valve as being attached to a rod forming a bucket- and plunger-pump, but it is obvious that the same valve may be differently employed by simply omitting that portion relating to the operation of driving. The pump from which this detail is taken has a plunger 20 inches in diameter, the bucket being 28 inches in diameter, and a stroke of 33 inches. This pump works against a head of 72 pounds per square inch. The revolutions of the engine were intended to be 18 per minute, but during the trial averaged but little more than 13 revolutions; the efficiency of the pump at that rate of speed was found to be 97.3 per cent.

FIG. 81.

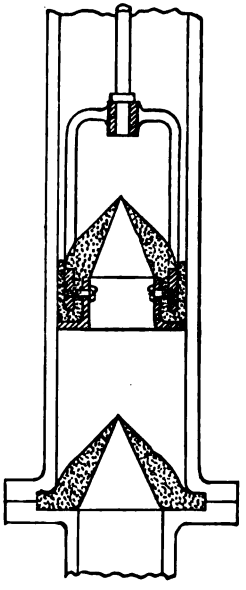


The indicator diagram, Fig. 81, taken from the above pump at the time of the trial, shows how nearly a large valve, and one of the above design, meets the exacting requirements of water-works service. The indicated agreed with the calculated

power required of the pump, which was found to be 47 horse-power. In the copying of the drawing the packing-box and ring at the top of the bucket was omitted for the sake of clearness.

Perreaux's Valve.—A novel form of valve is shown in Fig. 82; it is not used on this side of the Atlantic, and probably not extensively in Europe. The valve is made of india-rubber in the form of a tube flattened at one end, like the mouth-piece of a clarionet. The thickness of the sides of the upper part diminishes gradually to the top, where the two sides meet and form two lips, which, when the valve is in a state of rest, are in close contact and prevent the downward passage of the fluid. With any upward pressure the lips separate and allow of the upward passage of the fluid. The gradual diminution in thickness, or tapering of the sides forming the lips of the passage, enables the valve to open and close with the slightest variation of pressure, and, by properly proportioning, to resist any required amount of downward pressure. The passage for the fluid is larger in these valves than in any others of equal dimensions; they also possess the advantage of having a clear way, there being nothing whatever to retard the passage or flow of the fluid, and they close perfectly and instantaneously the moment the pressure from below ceases. The illustration shows two valves, one in the plunger and the other held between flanges forming the inlet-valve to the pump. The action is the same as in any ordinary pump: raising the piston producing an opening of the lower or suction-valve, whilst lowering it closes the valve. This valve is chiefly suitable for the raising and moving of

FIG. 82.

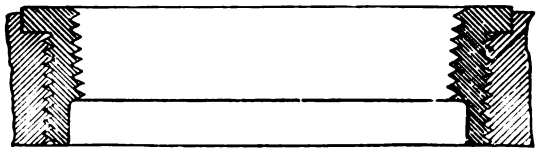


semi-fluid masses, like paper pulp, etc. These valves may be used singly for small pumps, or combined in sets for large pumps. But it may be remarked that, with the resistance of the material and the small dimensions of the orifices, an excess of motive-power is required, which may be an important element in large pumps. This valve is not to be recommended for heavy pressures; the ball-valve is believed to be much better for pressures greater than say 25 pounds per square inch.

These pumps have been made with rubber valves and piston, working in a glass barrel for handling acid.

Bushings for valve-seats are not often required except for repairs, but it sometimes happens that such a detail must be carried out in a new design. Fig 83 is a suggestion based

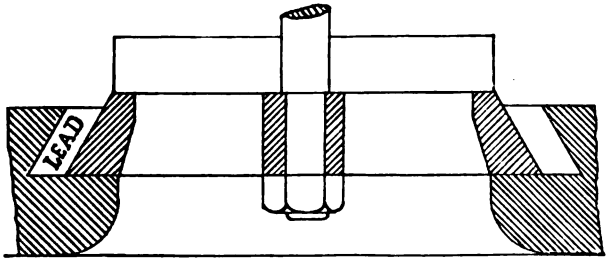
FIG. 83.



upon a successful experience. The cast-iron deck of the pump is reamed, counterbored, and tapped with a straight tap. The bushing is of gun-metal, threaded to correspond to the size of the tap, the collar being turned to fit the counterbored hole, and projecting slightly above the valve-deck of the pump to save the trouble of facing off any slight irregularities of the casting. The inside thread has the same taper as that of the valve-seat, say one inch to the foot. A steel driver made to fit the inside thread, with an adjustable screw-collar to tighten down upon the flat surface of the upper side of the bushing, makes a good tool for screwing these bushings in place. When the bushing is down to where it belongs, the adjustable screw-collar on the driver can be slackened and the driver easily and quickly withdrawn, after which the bushing is ready to receive the valve-seat.

Dovetail and Lead Joints for Valves.—Securing valve-seats in place by dovetail joints, calked with lead, as shown in Fig. 84, was a common practice a few years ago, when larger valves than are now commonly employed in large pumping engines were in general use. This arrangement is not well suited to valves less than 6 inches in diameter, and has but little to recommend it for valves of any size, although valve-

FIG. 84.

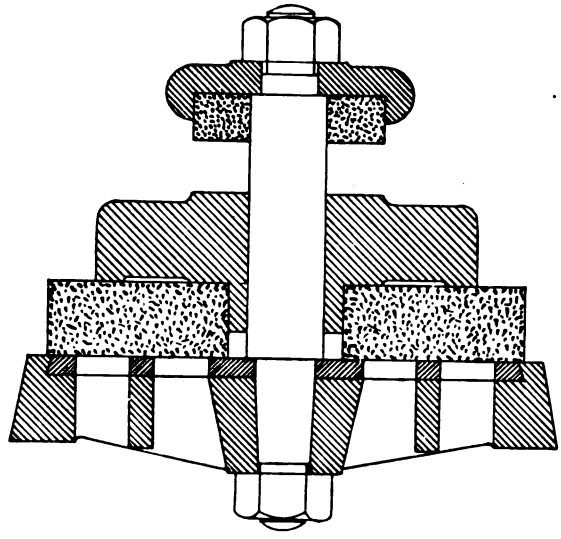


seats of 8 and 9 inches in diameter have been in use for many years having this kind of fastening. Now that the use of smaller valves which permit of screwed seats is the common practice, the above detail is not likely to be carried out in any important pumping engine contracts.

A weighted valve, such as that shown in Fig. 85, was much in use in this country twenty years ago for large pumping engines, but has been almost entirely displaced by the smaller valves with springs instead of weights. The valve-seat in the illustration is a cast-iron grid, faced with brass, the latter metal cast in a dovetail recess as shown, and afterwards faced off true to receive the valve. The spindle is of tough brass or Muntz-metal, fitted into a tapering hole in the seat. The weight is cast iron, and is made to slide loosely on the spindle, and to prevent noisy contact a small rubber buffer is included in the spindle-cap as shown. The valve-seat is held in place by lead calking, as shown in Fig. 84. Valves of this design, from 6 to 12 inches in diameter, are in use, but they

are somewhat sluggish in their movement and do not seat as quickly as similar valves furnished with spiral springs. They are only suited to large pumps and slow piston speeds.

FIG. 85.



Size of Valves.—It was formerly, say twenty-five years ago, the practice to fit water-ends with valves much larger in diameter than at present; valves from 5 to 8 inches in diameter being quite common in water-works practice.

A reaction afterwards set in, and the diameters of water-valves were then reduced so much that on some large pumping engines, built perhaps twelve or fifteen years ago, thin metal disks were employed less than 2 inches in diameter. This may be regarded as an error in judgment, which has since been very generally corrected, so that it may be said the common practice is now to confine the diameters of rubber disk-valves to between 3 and 4½ inches.

The area of clear water-way through a set of valves in a water-end should be not less than forty per cent. of the plunger area for pumps having a speed of 100 feet per minute; and if

that speed be increased to say 125 feet per minute, then the combined water areas through the valve-seats should be fifty per cent. of the plunger area; and in like manner 150 feet per minute would require sixty per cent. valve area; 175 feet per minute would require seventy-five per cent. valve area; and 200 feet per minute should have a valve area equal to the plunger area.

In order to get this valve area there is the temptation to make the valves as few in number and as large in diameter as possible, consistent with quiet action when the pump is working at its highest speed. It will be borne in mind that the two properties of the circle are to be considered when designing valves and seats. The area increases as the square of the diameter; the circumference varies directly as the diameter. The flow of water through the valve-seats has to do with the area only; the escape of the water under the valve into the chamber in which the water is to flow has to do with the circumference only.

Let us assume, by way of illustration, that a pump would require a single valve area, corresponding to 9 inches in diameter:

Area of 9 inches = 63.62 square inches.
Circumference of 9 inches = 28,27 inches.
Then 63.62 ÷ 28 27 = 2.25 inches lift required to make the circumferential opening equal to the area.

If, now, we try four valves, each 4½ inches in diameter, precisely the same area is had, but the circumference will be doubled, thus:

Area of 4½ inches = 15.90 × 4 valves = 63.60 inches total area.
Circumference of 4½ inches = 14.13 × 4 valves = 56.52 inches combined circumferences.
Then 63.60 ÷ 56.52 = 1.125 inches lift required of each valve to give a circumferential opening equal to the area, or one-half that required by the single valve.

It will be understood that both of the above illustrations have reference to the diameter of the opening, and not that

of the valves; to be exact, we must deduct the area of the grids, taking only the clear water-way through the seat, but the application would be the same.

TABLE III.

SHOWING THE PROPERTIES OF CIRCLES APPLICABLE TO VALVE CALCULATIONS.

Diameter. Inches.	Area. Square Inches.	Circumference. Inches.	Lift required to equal the Area. Inches.
1½	1.76	4.71	.375
2	3.14	6.28	.500
2½	4.91	7.85	.625
3	7.07	9.42	.750
3½	9.62	10.99	.875
4	12.57	12.56	1.000
4½	15.90	14.13	1.125
5	19.64	15.70	1.250
5½	23.76	17.27	1.375
6	28.27	18.84	1.500
6½	33.18	20.42	1.625
7	38.48	21.99	1.756
7½	44.18	23.56	1.875
8	50.26	25.13	2.000
8½	56.75	26.70	2.125
9	63.62	28.27	2.250
9½	70.88	29.84	2.375
10	78.54	31.41	2.500
10½	86.59	32.98	2.625
11	95.03	34.55	2.750
11½	103.86	36.12	2.875
12	113.09	37.69	3.000

The speed at which a pump will run noiselessly, assuming that an ample water-supply is had, will probably depend more on the relation of its valve area to that of its plunger area than almost anything else.

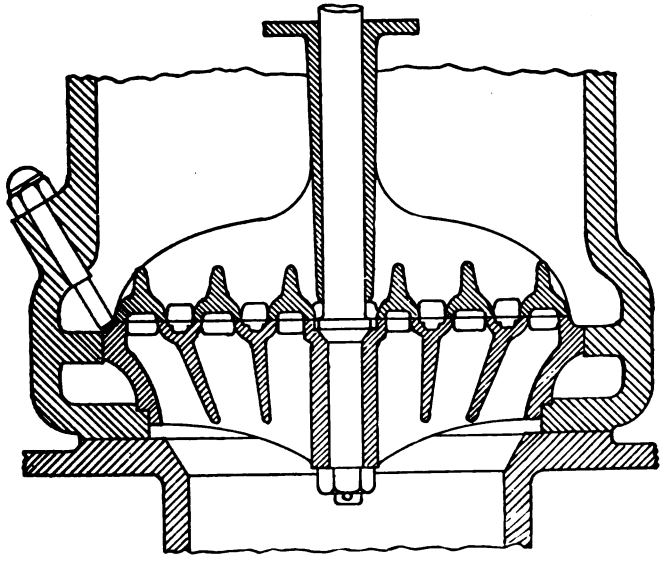
In a quick-running pump with too small a valve area an excessive lift is required of the valves, so that in the interval of seating a portion of the water in the pump-cylinder passes under the valves and back again into the suction-chamber; at the moment when the pressure overtakes the valves in their downward movement the velocity is so greatly accelerated as to force them violently down upon their seats, the pump becomes noisy, and nothing will relieve the pump but a reduc-

tion in the speed of plunger, suited to the proper and noiseless action of the valves.

Noisy action is not always confined to quick-running pumps; it is a common fault with nearly all low-priced pumps; the temptation evidently being to put in larger water-plungers than the valve area can supply at the common rating of 100 feet piston-speed per minute. The additional cost of a larger plunger in any properly-designed pump is a small one compared with that of an entire water-end, which would have to be supplied if a larger valve area must be furnished. The writer has known of pumps which barely measured 25 per cent. of valve area, when the least measurement should have been 40 per cent., a species of fraud rarely detected until too late to remedy it.

A concentric ring-valve is shown in Fig. 86; it has but recently been introduced in this country.

FIG. 86.



After a very exhaustive research in the matter of valve design, Professor Riedler, of Berlin, has given this the pref-

erence over all others for large valves, for quick-running engines. An inspection of the drawing will show that this design offers a large valve area for a comparatively small diameter of valve-chamber. This valve is intended to be operated by a suitable mechanism, which insures opening and closing at periods exactly corresponding to the changes in piston or plunger movement. This positive movement permits a higher piston speed than can be had in designs depending upon either gravity or springs for closing the valves. Its efficiency has certainly never been surpassed, and in point of simplicity, strength, and consequent durability it seems to leave but little more to be desired

Mechanically-operated water-valves are among the very earliest devices connected with the development of the direct-acting steam-pump. For one reason or another they never became popular in this country, and finally dropped out altogether, so that it was something of a surprise to American engineers when it was learned that several large, high-speed pumping engines had been built, and exhaustive experimental tests made, in Germany, in which mechanically-operated valves proved so highly satisfactory that their future adoption in all large and important high-speed pumping engines seems now to be nearly, if not altogether, certain.

Professor Riedler, of Berlin, has probably undertaken and accomplished more in the matter of indicator research in pumping engines than any other person now living. His designs for valves and valve-moving mechanism are the result of the thorough investigation of pumping machinery actually at work in mines and water-works stations. The valve-gear invented by him is the result of this investigation, and not the development of a chance thought. Large pumping engines made on his system are at work in Europe at piston-speeds of more than 300 feet per minute, and at high pressures such as usually attach to mining operations. Several designs, with descriptive matter, have been published by Professor Riedler in Germany, but so far as the writer knows they have never

been translated into English. It was the intention to have had translations made, in part, of Professor Riedler's papers, and certain engravings reproduced, but it was found that to make proper presentation would occupy more space than could be given the subject in this book. Whatever the writer may do in the future regarding this subject, he now refers the reader to:

I. *Indicator-Versuche an Pumpen* und Wasserhaltungs-Maschinen. Von A. Riedler, Professor des Maschinenbaues an der kgl. Technischen Hochschule in München. Mit 21 Tafeln und 24 Text Figuren. 1881.

II. Illustrated papers by Professor Riedler, published in the *Zeitschrift des Vereines Deutscher Ingenieure:*

Band xxvii., 1883, "Die unterirdischen Compound-Wasserhaltungs Maschinen am Mayrau-Schachte der Prager-Eisenindustrie-Gesellschaft zu Kladno."

Band xxix., 1885, "Constructions-Grundlagen der Pumpen- und Gebläse-Ventile."

Band xxxii., 1888, "Pumpen mit gesteuerten Ventilen."

Band xxxiv., 1890, "Neuere Wasserwerksmaschinen."

CHAPTER V.

AIR- AND VACUUM-CHAMBERS.

Air-Chambers.—All single pumps must be fitted with air-chambers if a smooth flowing and continuous delivery of water is desired. Duplex pumps have a discharge much more uniform than is the case in single pumps; but even they should have air-chambers to insure a perfectly uniform delivery.

If the discharge of water from a pump be intermittent, each successive impulse is equivalent to a blow; and this effect is not localized, but is distributed throughout the whole system. The final reaction, however, comes upon the pump, and if the pressure be a heavy one, leaky joints or broken castings are

FIG. 87.

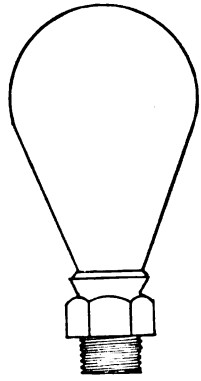
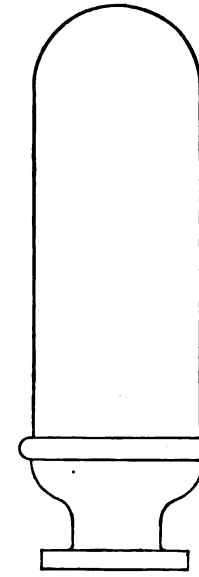


FIG. 88.



the almost inevitable result. A properly-designed air-chamber corrects this to a certain extent, but not wholly so. The largest air-chambers are required for single-acting pumps, and the smallest size for double-acting duplex pumps.

The form of an air-chamber admits of considerable latitude in design. For small pumps in which it is the common practice to use copper air-chambers, the design shown in Fig. 87 is seldom deviated from, because pump-makers with scarcely any exception buy them from certain coppersmiths who make their manufac-

FIG. 89.

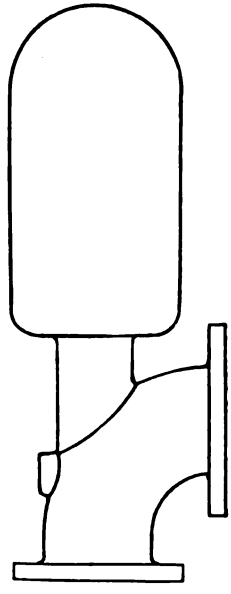
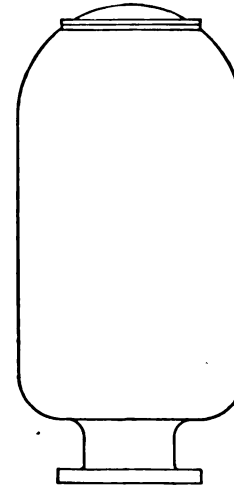


FIG. 90.



ture a leading specialty, and have adopted certain dimensions which meet all the ordinary requirements of the trade. These dimensions are approximately as below:

Diameter,	6	inches.	Height,	10	inches.	Base	threaded	for	1	inch	pipe tap.
"	8	"	"	14	"	"	"	"	1¼	"	"
"	9	"	"	15	"	"	"	"	1½	"	"
"	10	"	"	16	"	"	"	"	2	"	"

For highly-finished pumps the air-chambers are sometimes made of polished copper, riveted and soldered to a cast-iron or brass base, as in Fig. 88. The design of the base admits of a wide variety of treatment, but the whole does not materially differ in proportions from those of cast iron, except that they are usually smaller in diameter and made enough higher to get the proper cubic capacity.

Where larger air-chambers are required, it is customary to make them of cast iron, and usually of the form in Fig. 89 for medium-sized pumps, and Fig. 90 for large pumps. A table of dimensions and capacities suited to all ordinary requirements is here given:

TABLE IV.

ORDINARY DIMENSIONS AND CAPACITIES OF AIR-CHAMBERS.

Inside Measurement.		Capacity.		Remarks.
Diameter. Inches.	Height Inches.	Cubic Feet.	Gallons.	
12	24	1.57	11.75	The diameter of the neck and flange of an air-chamber will be governed somewhat by its intended location, but it will usually approximate one-third that of the air-chamber for the neck, with its corresponding diameter of flange.
14	28	2.49	18.64	
16	32	3.72	27.87	
18	36	5.30	39.66	
20	40	7.16	54.34	
22	44	9.68	72.48	
24	48	12.56	94.00	
26	52	15.98	119.42	
28	56	19.96	149.12	
30	60	24.55	183.60	
32	64	29.79	222.69	
34	68	35.76	266.59	
36	72	42.42	317.28	

Size of an Air-Chamber.—For single pumps, double-acting, the cubic contents of the air-chamber should be not less than three times that of the pump displacement; that is, a single stroke. This will be for moderate pressure and speed only, such as ordinary boiler-feeding, tank-service, etc. If for a pressure of 100 pounds per square inch and upwards, together with a very rapid piston movement, as in the case of

fire-pumps, then the air-chambers should be not less than six times the displacement of a single stroke of the pump-piston. For double-acting duplex pumps the cubic capacity of an air-chamber need not be more than one-half to two-thirds of the size given above. The position of the air-chamber should be on the highest portion of the pump, and always above the highest portion of the delivery opening.

The diameter of the neck of an air-chamber should be no larger than that necessary to give stability to it and insure proper strength of connection to the flange by which it is bolted to the pump. The larger the neck the greater will be the disturbance of the water-level in the air-chamber, and the less efficient will the air-chamber be in correcting the inequalities of flow.

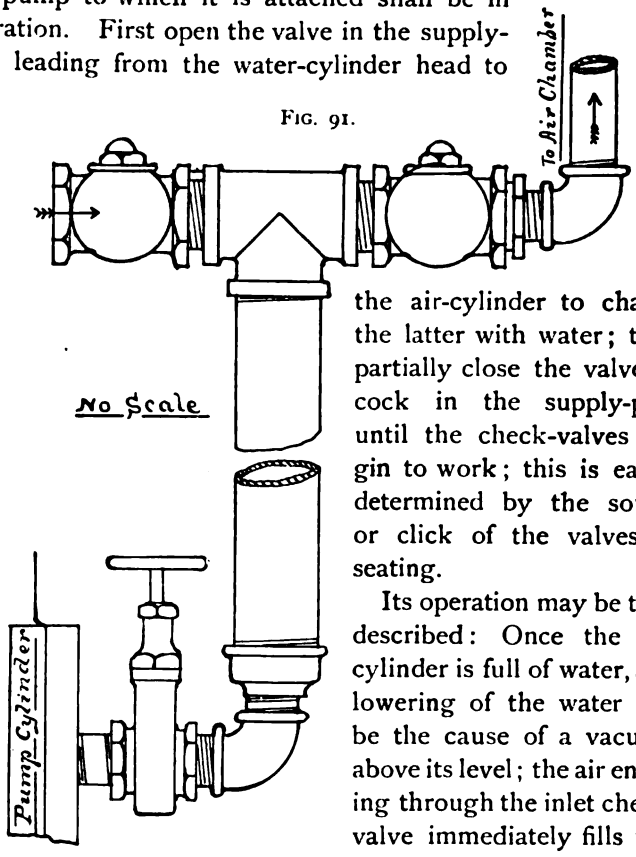
It is generally known that water contains air in suspension or solution, and in improperly-designed hydraulic machinery is likely to make trouble if there be air-pockets or cavities in which it may collect. Having this fact in mind one would naturally conclude that an air-chamber would always have its proper supply of air; that there would be no decrease in quantity at any time, even though it were not augmented by the liberation of air from the water when passing through the pump; but the reverse of this is true, and it is a common experience that air in large air-chambers almost wholly disappears, passing off, probably, with the water by absorption.

An automatic air-pump for supplying the air-chamber with air is shown in Fig. 91. It is exceedingly simple, and does its work in a very satisfactory manner. It consists of a piece of, say, 2½-inch wrought-iron pipe, about 30 inches long, which we shall name the air-cylinder, on the top of which is screwed a 2½-inch tee, one end having a 1¼-inch check-valve opening inward, and on the other end of the tee a ¾-inch check-valve, opening outward; a pipe from this small check-valve leads to the air-chamber. On the bottom of this 2½ inch pipe, which we have named the air-cylinder, is screwed a cap from which a 1-inch pipe, which we

shall name the supply-pipe, leads to the water-cylinder head; intermediate between the two is inserted a gate-valve or cock, which completes the whole apparatus, and which is suitable for pumps from 750,000 to 1,500,000 gallons in 24 hours.

To start this air-pump, it is necessary that the pump to which it is attached shall be in operation. First open the valve in the supply-pipe leading from the water-cylinder head to the air-cylinder to charge the latter with water; then partially close the valve or cock in the supply-pipe until the check-valves begin to work; this is easily determined by the sound or click of the valves in seating.

FIG. 91.



Its operation may be thus described: Once the air-cylinder is full of water, any lowering of the water will be the cause of a vacuum above its level; the air entering through the inlet check-valve immediately fills this space above the water at atmospheric pressure. The distance to which this water shall be lowered may be controlled by the valve or cock in the supply-pipe under the air-cylinder. The water is lowered in the air-cylinder by reason of its open connection with the water-cylinder of the pump, for each time the water-piston draws its

water the same vacuum is had in the air-cylinder as that in the suction-chamber of the pump. The air-cylinder being of so much larger area than the supply-pipe leading to the water-cylinder, there is not sufficient time for the air-cylinder to completely empty itself before the change in the direction of motion of the water-piston, which at once changes the direction of the flow in the supply-pipe, arresting the downward movement of water in the air-cylinder, forcing it upwards and filling the air-cylinder completely with water from the water-cylinder of the pump, thus driving the contained air in the air-cylinder through the delivery check-valve into the upper portion of the air-chamber. It will be seen that this operation is precisely similar to that of a reciprocating air-pump, the rise and fall of the water being precisely similar in effect to that of a reciprocating piston. There will be a greater velocity of water upwards in the air-cylinder than in its downward movement, because the downward movement will have a velocity due only to the pressure of the atmosphere, whereas the upward movement has a velocity due to the pressure given it by the action of the piston of the main pump. There will always be a little water carried over with the air into the top of the air-chamber; this excess of water serves a useful purpose in the fact that by it the clearance in the air-pump is absolutely eliminated.

There is no uniformity of opinion among managers of mines in general as to the utility of air-chambers on mining pumps, the drift of opinion being against them, mainly because they soon fill with water and become useless for the purpose intended. Owing to the indifference regarding this important detail of a pump, there has rarely been, until within a few years past, any device by which the pressure, and especially the volume of air, is maintained in the air-chamber, thereby failing to secure the benefits of an elastic cushion in the delivery-chamber of the pump. In the case of a single mine-pump a large air-chamber is of very great advantage, and under no circumstances should it be omitted.

If a duplex pump be used, the air-chamber is less essential to smooth working; but even then it should not be omitted.

VACUUM-CHAMBERS.

Definition.—Properly speaking, there is no such thing as a vacuum-chamber belonging to a pump; it is an air-vessel attached to the suction side of the pump to steady the flow. The use of a vacuum-chamber is precisely the reverse of that of the air-chamber, which has for its function the changing of an intermittent flow into a continuous one; whereas the vacuum-chamber has for its function the conversion of a continuous flow into an intermittent one.

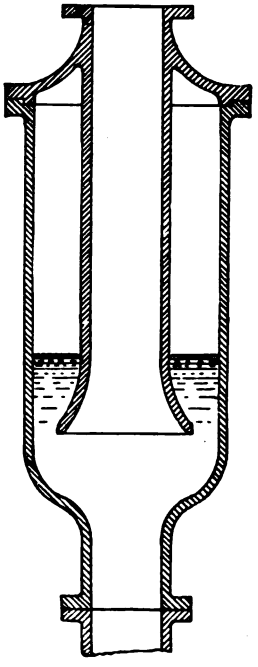
The flow of water into a pump is that due to the action of the atmosphere alone, and once the column of water is started its flow must be continuous if the best results are to be expected. The effect of the vacuum-chamber is to take away from the suction-chamber of the pump the water-hammer and other disturbing influences consequent upon a continuous flow into it, and from which the withdrawal of water is intermittent. The air in the vacuum-chamber forms an elastic cushion which will receive the excess of flow without noise, and give it out again as silently as it received it. The air is thus partially expanded and compressed at each wave or impulse of the water flowing into the pump-chamber.

The size of a vacuum-chamber need not ordinarily be more than one-half that of the air-chamber; a good practice is to make the cubic capacity twice that of a single displacement of a water-cylinder for a single pump. A less size would answer for a duplex pump, but should one be needed at all it ought to be a liberal one, and the above proportions will probably be none too large.

Suction-pipes are not always short and straight, but quite as likely to be long and crooked; it is for these latter that a vacuum-chamber becomes a necessity, and to get the best results it should be placed as near the pump as possible. The form of a vacuum-chamber should be similar to that of the

air-chamber; that is, it should have a much greater height than diameter. The table of air-chambers will give good proportions for vacuum-chambers also, and to which the reader is referred.

FIG. 92.



A good design for a vacuum-chamber is shown in Fig. 92. Its form permits of application immediately below the pump, and continue the suction-pipe down to the water-supply. There is nothing about the design that will interfere with its being made in any size, even for large water-works pumping engines. It consists of two castings flanged, faced, and bolted together, as shown in the engraving. The enlarged mouth-piece is advantageous in the fact that a full supply can be had without the formation of an eddy about the inner edge of the opening when the pump is working up to nearly the capacity of the suction-pipe.

The introduction of the suction-pipe into the air-vessel diminishes the volume of the latter to that extent; an increased height rather than diameter ought to be given it to make good the space thus occupied.

CHAPTER VI.

SUCTION- AND DELIVERY-PIPES.

Suction-pipes should be as short and direct as possible. In ordinary trade pumps the area of suction-pipes is approximately two-thirds that of the water-cylinder, but if the conditions are in any respect unusual, then the suction-pipe should nearly, if not entirely, equal that of the water-cylinder of the pump.

The suction side of the pump is subject to atmospheric conditions wholly, and everything which would tend to restrain the free flow of water into the pump should be carefully guarded against. It is not possible to dispense with angles and bends in a suction-pipe, but much of the increased resistance to the flow by reason of these bends can be overcome by increasing the size of the suction-pipe; this permits a slower movement of the water flowing towards the pump, so that the resistance is largely reduced.

The velocity of flow in a suction-pipe should not exceed 200 feet per minute. It is not an uncommon practice to make the suction-pipes for water-works pumps of the same diameter as that of the water-cylinders, and as 100 feet piston-speed per minute is the common speed for direct-acting pumps, it will be seen that the flow is a very moderate one indeed. In the specifications recently prepared by Mr. Freeman for the Underwriter Pump, the suction-pipes were made unusually large; for example, a

6 × 12 pump is to have a 6-inch suction-pipe.
7 × 12 " " 8 " "
9 × 12 " " 10 " "
10 × 12 " " 12 " "

These pumps are to be of the duplex type only, double-acting, and run at 70 "revolutions" per minute. By revolution is meant one complete circuit of the motion of any of the reciprocating parts of the pump, and for a duplex pump is equivalent to four single strokes; a somewhat unusual speed except for fire-pumps, as it is the equivalent to a piston-speed of 140 feet per minute.

The following table gives the relative proportions of water-cylinders to that of suction- and delivery-pipes as used by the writer with very satisfactory results:

TABLE V.

DIAMETERS SUITABLE FOR SUCTION- AND DELIVERY-PIPES FOR DUPLEX DIRECT-ACTING PUMPS RUNNING AT 100 FEET PISTON-SPEED PER MINUTE.

Water-Cylinder.		Suction-Pipe.			Delivery-Pipe.		
Diameter.	Area.	Diameter.	Area.	Velocity of Flow at 100 Feet.	Diameter.	Area.	Velocity of Flow at 100 Feet.
Inches.		Inches.		Feet.	Inches.		Feet.
4	12.57	3	7.07	178	2	3.14	400
5	19.64	4	12.57	156	3	7.07	277
6	28.27	5	19.64	143	4	12.57	224
7	38.48	6	28.27	136	5	19.64	196
8	50.27	6	28.27	180	5	19.64	256
9	63.62	8	50.27	126	6	28.27	225
10	78.54	8	50.27	156	7	38.48	204
12	113.09	10	78.54	144	8	50.27	224
14	153.93	12	113.09	136	10	78.54	196

Suction-Pipes should be of One Diameter only from End to End.—Enlargements are always to be avoided, because they interfere with that uniform rate of flow so essential to the proper filling of a pump. The pipe should have a continuous rise from the water-level of the source of supply to the pump; any irregularity in the laying which would result in the formation of an air-pocket in the pipe must be corrected if good results are to be expected.

If the suction-pipe is larger in diameter than the opening into the pump, a short conical piece of pipe, one end fitting

the pump and the other end fitting the suction-pipe, should be made and attached either to the pump or as close to it as possible.

Suction-pipes must be tight, absolutely tight; about this there must be no mistake, as anything short of it means uncertainty and loss of efficiency in working, if not a complete failure of the pump to perform the service for which it was intended.

A leaky suction-pipe must be tested joint by joint until the leak is discovered and corrected. If the pipe has been laid in a trench and covered over with earth, it may seem like

FIG. 93.

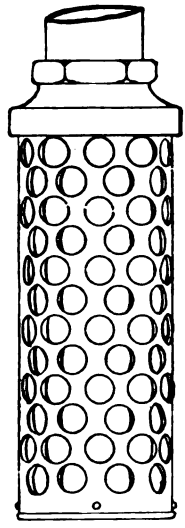
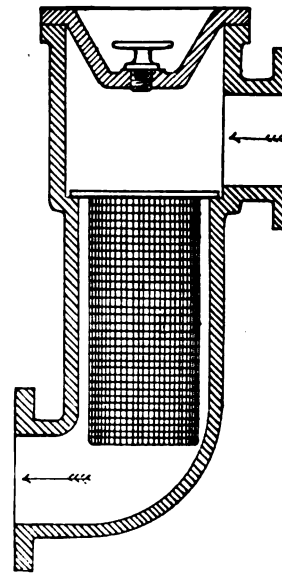


FIG. 94.

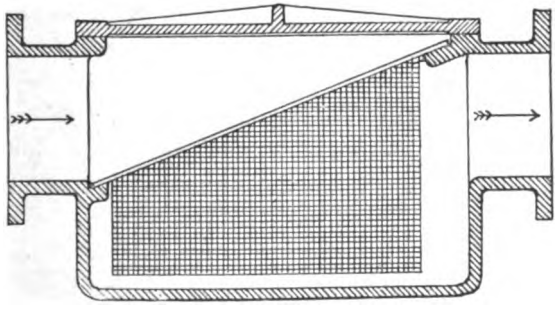


a great deal of labor and expense to lay it bare, but there is no other way out of it. A practical method for testing a suction-pipe is to put a blank flange over the lower joint in the water, a similar one on the delivery-flange of the pump,

fill the suction-pipe and pump with water, insert in any convenient place in the suction-pipe a pressure-gauge, then with a small hand force-pump, unless another source of pressure is had, get up a pressure of say 50 to 60 pounds per square inch, and then shut off the force-pump connection with this pressure on the suction-pipe and pump. Now watch the pointer on the dial of the pressure-gauge, and if it loses pressure it is certain that a leak exists somewhere; nothing now remains but to search it out and fix it.

If the suction-pipe is not in a covered trench, but accessible throughout its length to the water-supply, a leak can readily

FIG. 95.

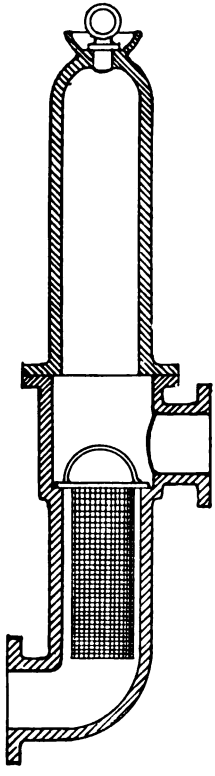


be discovered when the pump is in use by the sound of the air rushing into the pipe; having thus located the joint, for that is where the leak is likely to occur, the exact location of the hole or leak can generally be found by means of a lighted candle, the in-rushing air diverting the flame towards the hole in the pipe, if held near enough to be so influenced. Suction-pipes should be provided with a strainer, to keep out floating matter, such as twigs, leaves, fish, etc., if the supply be had from a stream or a pond.

A strainer may be made in any one of a variety of ways, and answer the purpose for which it is intended. A wooden or iron skeleton framing covered with wire-cloth of say ½-inch or ⅝-inch mesh, carefully fitted around the bottom of the

suction-pipe and securely fixed in the stream, makes a good strainer. A simple and effective strainer for small pipes is shown in Fig. 93. It is nothing more than a thin pipe, enough larger in diameter than the suction-pipe as to present a sufficient area of opening through the holes in its outer surface. There should be no holes in the bottom. The combined area of these holes should in no case be less than three times the area of the suction-pipe; and if the holes are likely to be covered by leaves, the area should be four times that of the suction-pipe. A vertical or slanting side to a strainer is always to be preferred, because it is less likely to retain leaves, etc., on the strainer after the flow of water had ceased at the stoppage of the pump.

FIG. 96.



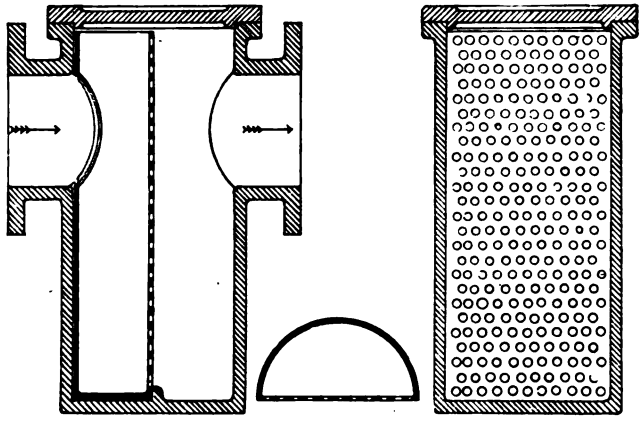
It is frequently the case that a strainer cannot be conveniently reached for cleaning if attached to the end of the suction-pipe. There would, then, be an advantage in placing it near or attaching it to the pump. The arrangement shown in Fig. 94 is a good one for small pipes, but for large pipes the strainer should be horizontal, and may be placed at any convenient location between the source of water-supply and the pump. Such a strainer is illustrated in Fig. 95.

A combined strainer and vacuum-chamber is shown in Fig. 96. Like the former, the strainer-basket inside can be readily taken out and cleaned. As the outlet attaches directly to the pump, no better place could be selected for the vacuum-chamber than the top flange of the side-pipe which holds the strainer; this chamber is provided at the top with a screw plug, so that, should it be necessary to charge the pump with water, as is often the case with long suction-pipes not

wholly tight, it can easily be done through the opening thus provided.

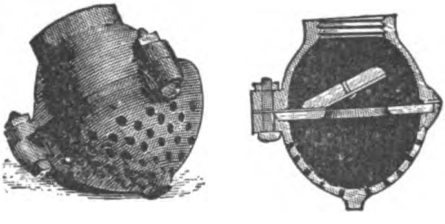
The strainer shown in Fig. 97 has a semicylindric vessel located on one side of the side-pipe. Holes are drilled

FIG. 97.



through the flat side extending across the diameter of the side-pipe; any floating matter which will not pass through the holes collects in the strainer-vessel and can be quickly removed. This is a more substantial arrangement than those illustrated above, but it is more expensive to make, because a larger diameter of side-pipe is necessary to get the same area of opening that could be secured through the meshes of a wire-cloth basket.

FIG. 98.

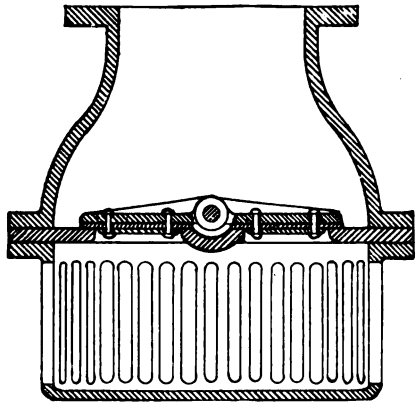


A foot-valve should be attached to all suction-pipes whenever of unusual length, or lift of water. In selecting a valve, be sure that a clear water-way is had through the seat of at least the area of the pipe with which it is to be used. Fig. 98

is the common design, as sold in all supply stores, for combined foot-valves and strainers in all sizes up to 4-inch wrought-iron pipe.

A sectional elevation of a common form of butterfly-valve is shown in Fig. 99. Foot-valves of this type are usually fitted with leather-faced clack-valves seating on a tool-finished cast-iron bottom or seat. The hinged valves are usually secured by the same centre-pin, or through-going bolt, stops being provided so that when a full water-way is had the valves cannot fall back, but must return, each to its own seat.

FIG. 99.



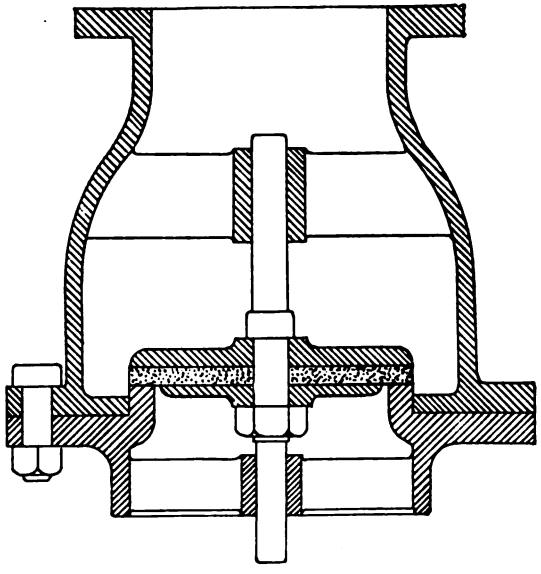
Foot-valve castings should be amply strong, so that any delivery leakage past the valves in the pump into the suction-pipe would not split the foot-valve.

On large water-works valves it is quite a common practice to put a safety- or relief-valve on the side of the foot-valve casing or shell; this relief-valve being loaded or set for a pressure less than that which would endanger either the foot-valve or the suction-pipe, both of which are generally made lighter than pressure pipes or valves for the same size or diameter. This relief-valve is not intended to be anything else than simply a leak in the suction-pipe, set to a certain known pressure, to prevent an accumulation of a higher pressure dangerous to the suction-pipe or foot-valve. This relief-valve need not be a large one; an inch-and-a-half valve will answer for an 8- or 10-inch pipe, and a two-inch valve for a 12- or 14-inch pipe.

A foot-valve with a single india-rubber disk held between metal plates is shown in Fig. 100. This design is suitable for

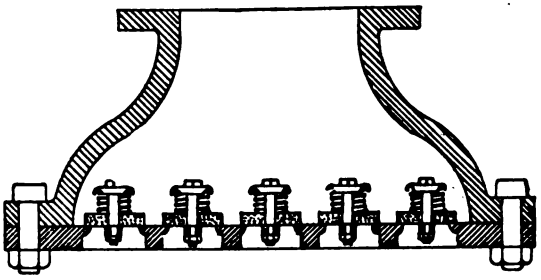
pipes up to 8 or 10 inches in diameter. The drawing shows a spindle extending both above and below the valve passing through bored guides. This is a method of construction which the writer does not wholly favor, as it is difficult to get the upper and lower holes exactly in line with each other; but for a foot-valve having a rubber face, as shown in the engraving, the fitting need not be very exact, so that play enough can be had in the guides to allow the valve to seat without any binding stress on the spindle.

FIG. 100.



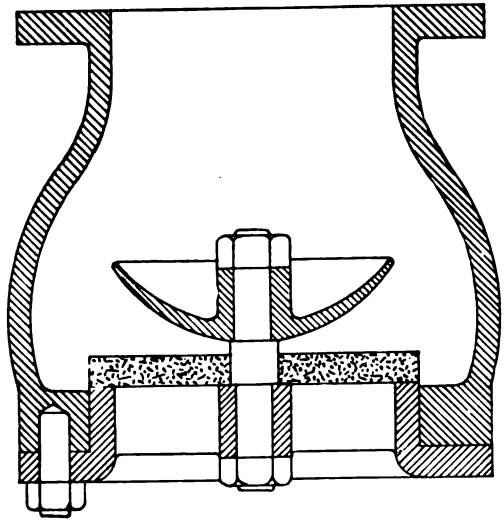
The design shown in Fig. 101 consists of a number of small valves similar in construction to those used in the water-end of a pump. This design is not in so common use as the valve shown in Fig. 99, nor does the writer regard it as favorably. Foot-valves

FIG. 101.



should preferably be so designed that the seating would be by gravity alone, and wholly unassisted by springs. The action of a foot-valve is different from that of a pump-valve, which alternately rises and falls at each stroke of the plunger. In a properly-designed pump the foot-valve ought not close from the moment the pump starts until it is stopped, and as pumps are often in continuous operation for several hours or days, the springs, if any, would be under a tension, the extent of which would be measured by the lift of the valve.

FIG. 102.



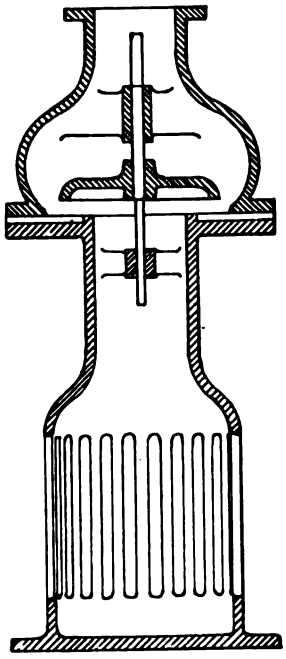
By the use of a clack- or disk-valve seating by gravity no such effects can follow, no matter to what height of lift the valve may be subjected, and is wholly uninfluenced as to time.

The writer does not recommend the use of a single disk of india-rubber, as illustrated in Fig. 102, for use as a foot-valve, because the flow of the water being continuous and always in the same direction, the valve is forced upwards against the curved guard, and is likely to receive a permanent set or dishing, and is in consequence less likely to fit the flat seat tight enough to prevent leakage around its outer edge.

The combined foot-valve and strainer shown in Fig. 103 is from a design executed for the water-works at La Chaux de Fonds. The flat plate on the bottom of the

strainer rests upon a stone foundation. The upper flange contains the rubber seating upon which the metal valve rests when the pump is not in operation. This valve has a central spindle passing through bored guides, one each in the casting above and that below the valve, a method of construction not wholly free of objections. In order to insure perfect alignment the upper and lower flanges should, after the proper adjustment of each other to the valve, be further fitted with dowel-pins, in order that the upper and lower guides may be exactly brought in line, if for any reason it should be necessary to take them apart.

FIG. 103.



Delivery-pipes need not be more than one-half the area of the water-cylinder for any ordinary service. The movement of the water in the delivery being subject to that of the pump by which it is forced, and not to atmospheric conditions, makes the problem one of friction only, and this amounts to so small a fraction of the total friction that it is seldom taken into account. In fact, it is quite an unusual thing to ascertain experimentally what the friction-loss in the water-end of a pump really is. The allowance usually made to cover the friction-loss in the distributing system is large enough to include that of the pump also.

The sizes for delivery-pipes as given in Table V. are those in very common use, and as the permissible velocity of discharge is 400 feet per minute, it will be seen that for all except the first size the dimensions are very liberal. For use in a factory system the diameters of the delivery-pipes in all pumps

TABLE VI.

FRICTION-LOSS IN POUNDS PRESSURE

For each 100 Feet of Length in Different Size clean Iron Pipes discharging Given Quantities of Water per Minute. Also Velocity of Flow in Pipe in Feet per Second.—*G. A. Ellis, C.E.*

Gallons discharged per Minute.	½-Inch.		¾-Inch.		1-Inch.		1¼-Inch.		1½-Inch.		2-Inch.		2½-Inch.		3-Inch.		4-Inch.		6-Inch.		Gallons discharged per Minute.
	Velocity in Pipe per Second.	Friction-Loss in Pounds.	Velocity in Pipe per Second.	Friction-Loss in Pounds.	Velocity in Pipe per Second.	Friction-Loss in Pounds.	Velocity in Pipe per Second.	Friction-Loss in Pounds.	Velocity in Pipe per Second.	Friction-Loss in Pounds.	Velocity in Pipe per Second.	Friction-Loss in Pounds.	Velocity in Pipe per Second.	Friction-Loss in Pounds.	Velocity in Pipe per Second.	Friction-Loss in Pounds.	Velocity in Pipe per Second.	Friction-Loss in Pounds.	Velocity in Pipe per Second.	Friction-Loss in Pounds.	
5	8.17	24.6	3.63	3.3	2.04	0.84	1.31	0.31	0.91	0.12	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	5
10	16.3	96.0	7.25	13.0	4.08	3.16	2.61	1.05	1.82	0.47	1.02	0.12	. .	. .	. .	. .	. .	. .	. .	. .	10
15	. .	. .	10.9	28.7	6.13	6.98	3.92	2.38	2.73	0.97	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	15
20	. .	. .	14.5	50.4	8.17	12.3	5.22	4.07	3.63	1.66	2.04	0.42	. .	. .	. .	. .	. .	. .	. .	. .	20
25	. .	. .	18.1	78.0	10.2	19.0	6.53	6.4	4.54	2.62	. .	. .	1.63	0.21	1.13	0.10	. .	. .	. .	. .	25
30	. .	. .	. .	. .	12.3	27.5	7.84	9.15	5.45	3.75	3.06	0.91	. .	. .	. .	. .	. .	. .	. .	. .	30
35	. .	. .	. .	. .	14.3	37.0	9.14	12.4	6.36	5.05	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	35
40	. .	. .	. .	. .	16.3	48.0	10.4	16.1	7.26	6.52	4.09	1.6	. .	. .	. .	. .	. .	. .	. .	. .	40
45	. .	. .	. .	. .	. .	. .	11.7	20.2	8.17	8.15	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	45
50	. .	. .	. .	. .	. .	. .	13.1	24.9	9.08	10.0	5.11	6.44	3.26	0.81	2.27	0.35	1.28	0.09	. .	. .	50
75	. .	. .	. .	. .	. .	. .	19.6	56.1	13.6	22.4	7.66	5.32	4.9	1.8	3.4	0.74	. .	. .	. .	. .	75
100	. .	. .	. .	. .	. .	. .	. .	. .	18.2	39.0	10.2	9.46	6.53	3.2	4.54	1.31	2.55	0.33	1.13	0.05	100
125	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	12.8	14.9	8.16	4.89	5.67	1.99	. .	. .	. .	. .	125
150	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	15.3	21.2	9.8	7.0	6.81	2.85	3.83	0.69	1.7	0.1	150
175	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	17.1	28.1	11.4	9.46	7.94	3.85	. .	. .	. .	. .	175
200	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	20.4	37.5	13.1	12.48	9.08	5.02	5.11	1.22	2.27	0.17	200
250	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	16.3	19.66	11.3	7.76	6.39	1.89	2.84	0.26	250
300	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	19.6	28.06	13.6	11.2	7.66	2.66	3.4	0.37	300
350	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	15.9	15.2	8.94	3.65	3.97	0.5	350
400	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	18.2	19.5	10.2	4.73	4.54	0.65	400
450	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	20.4	25.0	11.5	6.01	5.11	0.81	450
500	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	22.7	30.8	12.8	7.43	5.67	0.96	500

TABLE VI.—*Continued.*

FRICTION-LOSS IN POUNDS PRESSURE

For each 100 Feet of Length in Different Size clean Iron Pipes discharging Given Quantities of Water per Minute Also Velocity of Flow in Pipe in Feet per Second.—*G. A. Ellis, C.E.*

Gallons discharged per Minute.	8-Inch.		10-Inch.		12-Inch.		14-Inch.		16-Inch.		18-Inch.		20-Inch.		24-Inch.		30-Inch.		Gallons discharged per Minute.
	Velocity in Pipe per Second.	Friction Loss in Pounds.	Velocity in Pipe per Second.	Friction-Loss in Pounds.	Velocity in Pipe per Second.	Friction-Loss in Pounds.	Velocity in Pipe per Second.	Friction-Loss in Pounds.	Velocity in Pipe per Second.	Friction-Loss in Pounds.	Velocity in Pipe per Second.	Friction-Loss in Pounds.	Velocity in Pipe per Second.	Friction-Loss in Pounds.	Velocity in Pipe per Second.	Friction-Loss in Pounds.	Velocity in Pipe per Second.	Friction-Loss in Pounds.	
250	1.59	0.07	1.02	0.03	0.71	0.01	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	250
500	3.19	0.25	2.04	0.09	1.42	0.04	1.04	0.017	0.8	0.009	0.63	0.005	. .	. .	. .	. .	. .	. .	500
750	4.79	0.53	3.06	0.18	2.13	0.08	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	750
1,000	6.38	0.94	4.08	0.32	2.84	0.13	2.08	0.062	1.6	0.036	1.26	0.02	1.02	0.012	0.72	0.005	0.45	0.002	1,000
1,250	7.97	1.46	5.1	0.49	3.55	0.2	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	1,250
1,500	9.57	2.09	6 12	0.7	4.26	0.29	3.13	0.135	2.39	0.071	1.89	0.04	. .	. .	. .	. .	. .	. .	1,500
1,750	. .	. .	7.15	0.95	4.96	0 38	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	1,750
2,000	. .	. .	8.17	1.23	5.67	0.49	4.17	0.234	3.19	0.123	2.52	0.071	2.04	0.042	1 44	0.02	0.91	0.006	2,000
2,250	. .	. .	. .	. .	6.38	0.63	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	2,250
2,500	. .	. .	. .	. .	7.09	0.77	5.21	0.362	3.99	0.188	3.15	0.107	. .	. .	. .	. .	. .	. .	2,500
3,000	. .	. .	. .	. .	8.51	1.11	6.25	0.515	4.79	0.267	3.78	0.15	3.06	0.091	2.16	0.047	1.36	0.012	3,000
3,500	. .	. .	. .	. .	. .	. .	7.29	0.697	5.59	0.365	4.41	0.204	. .	. .	. .	. .	. .	. .	3,500
4,000	. .	. .	. .	. .	. .	. .	8.34	0.91	6.38	0.472	5.04	0.263	4.08	0.158	2.88	0.067	1.82	0.022	4,000
4,500	. .	. .	. .	. .	. .	. .	. .	. .	7.18	0.593	5.67	0.333	. .	. .	. .	. .	. .	. .	4,500
5,000	. .	. .	. .	. .	. .	. .	. .	. .	7.98	0.73	6.3	0.408	5.11	0.244	3.6	0.102	2.27	0.035	5,000
6,000	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	7.56	0.585	6.13	0.348	4.32	0.146	2.72	0.048	6,000
7,000	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	7.15	0.472	5.04	0.196	3.18	0.065	7,000
8,000	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	8.17	0.612	5.76	0.255	3.63	0.083	8,000
9,000	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	6.47	0.323	4.08	0.105	9,000
10,000	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	. .	7.19	0.398	4.54	0.131	10,000

having water-cylinders 6 to 14 inches diameter, both inclusive, could safely be reduced, beginning with a 3½-inch delivery-pipe on the 6-inch pump, and ending with an 8-inch delivery-pipe for the 14-inch pump.

Duplex pumps with water-cylinders 8, 10, 12, and 14 inches diameter are in common use in small water-works, and there should be no reduction in the diameters of the delivery-pipes for such a service.

How far vertically and horizontally can water be sucked at a definite speed by a common bucket- or ram-pump? This question was answered by Mr. James McCreath in a paper read before the Mining Institute of Scotland in 1883, from which the following abstract is made:

The sucking of water being dependent on the pressure of air, we may, for all practical purposes, consider air as a perfect gas. The pressure which the atmosphere exerts is never much less, at the level of the sea, than will support a column of water 32 feet high, and for every 262 feet above sea-level the pressure is about one-hundredth of itself less, according to Rankine.

The proportion of the height of 32 feet to which water can be raised in a pump depends upon the proportion of air within the pump which the pump can extract, and this depends upon its proportions. For example: A pump the bucket of which at top of stroke is 20 feet above the level of the water, the length of stroke being 2 feet and the clack being 16 feet above the water, can never suck the water up to the clack, because at the bottom of the stroke the air-pressure must be slightly more than that of the atmosphere, and at the top of the stroke, the air there occupying double the space, the pressure must be slightly more than half of that, and therefore the water will not rise to quite the half of 32 feet, which it would require to do to reach the clack. If the pump be filled with water from the outside, it will raise water until air again collects in it.

The following are examples of what the utmost rarefaction of air producible by various lengths of stroke and distances

between bucket and clack can do, 32 feet of water being taken as equivalent to the atmospheric pressure:

WITH THE LOW CLACK.

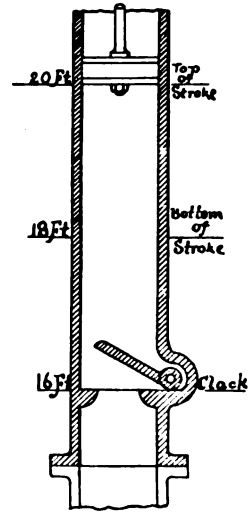
Length of Stroke in Feet.	Greatest Height of Bucket at Top of Stroke above Water in Well which will admit of Water being raised.
$1 \times 32 = 32$	$2 \sqrt{32} = 11.3$ feet.
$2 \times 32 = 64$	$2 \sqrt{64} = 16$ "

WITH THE HIGH CLACK.

Length of Stroke in Feet.	Distance between Bucket and Clack-Valves at Bottom of Stroke.	Height Water will raise to Clack.
1	1	$\frac{1}{2} \times 32 = 16$
2	1	$\frac{2}{3} \times 32 = 21\frac{1}{3}$

This raises water only to the clack. Very little of a longer stroke would raise it to the bucket, for the air-space to be exhausted lessens as the water rises above the clack; but some allowance must be made for the water returning through the valve while it is closing, provided a sufficient pause at end of stroke be not made to let it fall.

FIG. 104.



If the water get above the bucket, either by suction or by being filled from the outside, and the bucket be steadily raised, the water will in time follow the bucket until it reaches 32 feet above the water in the well; but if it does not reach the bucket, of course no water will be pumped.

The height to which water will rise in a ram-pump may be calculated in the same way, the space under and around the ram being reduced to an equivalent length of pipe of the diameter of the ram between the bucket and the clack. Provision ought also to be made for the escape of air remaining around the ram above the level of the exit-pipe.

It will thus be seen that the capacity of an ordinary pump to clear itself of air depends directly upon the length of the stroke, and, within certain limits, inversely as the distance between the bucket and the clack.

Weisbach gives the following rules obtained by experiment for the loss of energy of flowing water:

1. When flowing through short tubes whose diameter is equal to their length, the water is contracted to 0.61 of the area of the tube.

2. When passing round an elbow at right angles, 0.984 of (or very nearly the whole) head corresponding to the velocity is lost.

3. When passing round a circular bend, whose radius of curvature is to the radius of the cross-section of the pipe as 10 is to 4, the loss of head is equal to about $\frac{1}{5}$ of that corresponding to the velocity.

4. When passing through a valve similar to that shown in Fig. 104, where the cross-section of the aperture to that of the pipe is 0.535, the co-efficients of resistance are as follows:

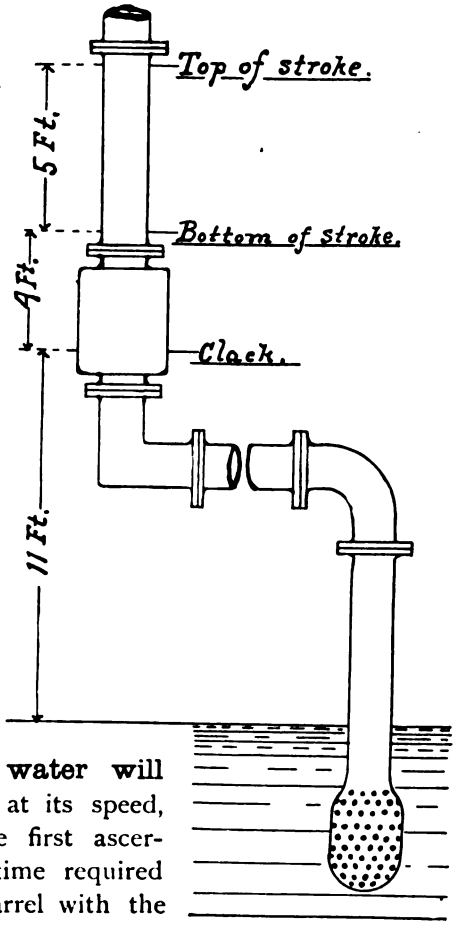
Angle of opening	15°	20°	25°	30°	35°		
Co-efficients of resistance . .	90	62	42	30	20		
Angle of opening	40°	45°	50°	55°	60°	65°	70°
Co-efficients of resistance . .	14	9.5	6.6	4.6	3.2	2.3	1.7

Let the foregoing rules be applied to a bucket-pump having bends under the clack, as shown in Fig. 105, the length of pipe from the bottom of the suction-pipe to the bucket at half-stroke = 100 feet, the height of the bucket at the top of the stroke above the surface of the water in the well = 20 feet, and above the clack at the bottom of the stroke = 4 feet, the stroke = 5 feet, and speed required, 12 strokes a minute, the pump being driven by a crank whose rotating motion is uniform.

In order to find whether a pump can clear itself of air, assume that the weight of the bucket-valve be such as to

reduce the effective stroke for air to 4¾ feet, and the weight of the clack-valve and the slip of water past the valve to take 6 inches from the column of water capable of being raised. The length of the stroke, 4¾ feet divided by 9 feet, the distance between the clack and the bucket at top of stroke, gives extraction of $\frac{4\frac{3}{4}}{9} \times 32 = 17$ feet; and deducting 6 inches for weight of clack-valve and for slip, leaves 16 feet 6 inches which the water would rise to the clack. But the bucket at bottom of stroke is only 15 feet above the water in the well, and there is, therefore, a large margin to cover imperfections in the pump.

FIG. 105.



In order to ascertain whether the water will follow the bucket at its speed, let the resistances be first ascertained, and then the time required to fill the working-barrel with the remaining pressure.

The mean velocity of the pump assumed is 2 feet per second, and the resistances at this velocity are,—

	Loss of Head in Feet.
Snore-holes and turn into pipe when sum of area of snore-holes = area of suction of pipe $\left\} \frac{\left(2 \times \frac{100}{61}\right)^2}{64.4}\right.$	= 0.167
Circular bend . . . . . $\frac{2^2}{64.4} \times \frac{1}{8}$	= 0.012
Square elbow . . . . . $\frac{2^2}{64.4}$	= 0.062
Valve at 20° . . . . . $\frac{2^2}{64.4} \times 62$	= 3.857
Friction of the water in the pipes, say of 12 inches diameter	= 0.165
Total resistance due to square of mean velocity	= 4.263

Weisbach shows that the resistance is as the mean square of velocity, and that in a uniformly accelerated or uniformly retarded motion this is twice the square of the mean velocity, and in a crank motion 1.645 times the square of the mean velocity. Assume, meanwhile, that the water follows the bucket, the total resistance will be equivalent to 4.2635×1.645, equivalent to a head of about 7 feet. The bucket at bottom of stroke being 15 feet above the water in the well, the pressure at first for raising the water and overcoming resistances is equivalent to 17 feet; but deducting 7 feet for resistances, the pressure available for giving motion to the water will be only 10 feet.

Taking the following approximate formula for time of filling the working-barrel (with the acceleration modified), suitable where the extremes of acceleration do not vary greatly, we have,—

$$\text{Time} = 2\,\frac{\sqrt{\text{height}}}{\sqrt{\text{twice the mean acceleration}}} = 2\sqrt{\frac{5}{64.4 \times \frac{10 - 2.5}{100}}} = 2.034 \text{ seconds.}$$

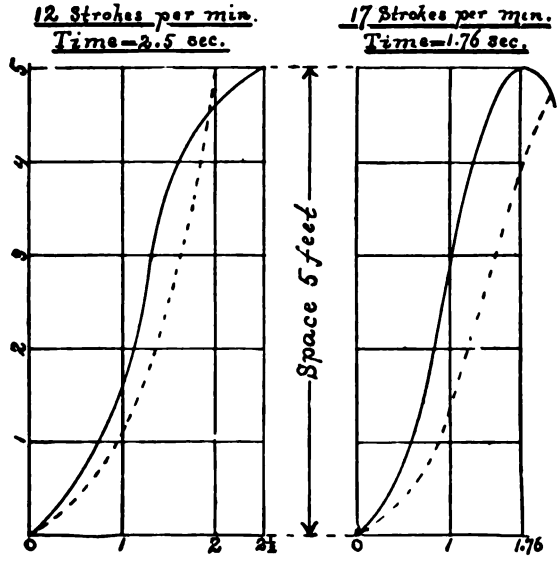
But the time the bucket takes to make the half-stroke is 2½ seconds, so that there is nearly half a second to cover the imperfections of the pump and any hinderance there may be at the beginning of the stroke from the interference of the bucket with the motion of the water. If an attempt be made to drive the pump at 17 strokes, the water will lag behind the

bucket, and they will meet on the return stroke, the water moving with a velocity of about 4.8 feet per second and the bucket with a velocity of about 1.66 feet per second, together about 6.46 feet per second.

If the water were so far behind the bucket as to meet it at half-stroke, a much more serious shock would arise, because in addition to the momentum of the water the momentum of machinery at a higher velocity, backed by the steam-pressure, would have to be met.

The diagram, Fig. 106, will show the relative motions of the bucket and the water, supposing the average resistance to

FIG. 106.



be uniform. The speed of the water would, however, if unretarded, be actually greater at the beginning of the stroke, and it will be less at the middle of the stroke; the top will therefore be reached a little later than shown.

The foregoing calculations show that the largest resistance is due to the clack-valve, and therefore special attention requires to be given to find a valve giving little resistance

when either the height or distance which the water has to be sucked is such as to raise a doubt of the pump working well.

What would be still more effective would be to double the area of the pump and drive at half the speed, and so reduce both the resistances and the work lost in the motion of the water to a fourth. With the pump already instanced the same quantity of water could be sucked 680 feet as freely with a 17-inch bucket and pipes going 6 strokes a minute as could be done 100 feet with a 12-inch bucket and pipes going 12 strokes a minute; and if, in addition, the stroke could be lengthened downwards 4 feet, retaining the same velocity, the water could easily be drawn 1415 feet.

The calculation of pressure caused by a shock in pumps may be obtained as follows: Calculate the amount of work in compressing the water and extending the pipe, which would be equivalent to the work due to the fall of the water, keeping in view that a load suddenly applied to an elastic substance will produce double the strain that the same load would do if applied gradually.

This may be illustrated in the case of a spring-balance and a half-pound weight. If the weight be put suddenly in the balance it drops down until the index shows nearly a pound of pressure, and then rebounds, and, as it oscillates alternately above and below the position of the half a pound of steady load, gives a very good illustration of what is called a "live load."

Example from Actual Experience.—The height of a column of water is 339 feet; diameter of pipe, 26 inches. A bucket-piece was broken by the water being suddenly stopped when descending with a velocity of 5 feet per second. Thickness of pipe, 1¾ inches. We may take the compressibility of water at $\frac{1}{294,000}$ of itself for each pound of steady load; extensibility of iron $\frac{1}{17,000,000}$ of itself for each pound of steady load. Weight of water = 147 pounds per square inch.

Let it be assumed first that the work due to the fall has to be met wholly by the compression of the water, then

$$\left.\begin{array}{l}\text{The work due to velocity of 5 feet} \\ \frac{V^2}{2g} \times \text{weight of 1 foot of water}\end{array}\right\} = \text{half of ultimate pressure} \times \text{amount of compression.}$$

$$\frac{25}{64.4} \times .433 = \frac{x}{2} \times \frac{x}{294{,}000}$$

$x = 314$ pounds per square inch.

This is the ultimate pressure required to meet the momentum of the water if moving horizontally, but the column above referred to being vertical, it has upon contact the additional compressing force due to a sudden application of a load of 147 pounds,—viz.:

Double the weight applied	= 294	pounds.
Add pressure due to stopping 5 feet velocity	= 314	"
Total pressure if work be met only by water	= 608	"

The work, however, is partially met by the stretching of the iron pipe, and the proportions borne by each are:

$$\underset{\text{Work on Water.}}{26'' \times 26'' \times \frac{3.1416}{4} \times \frac{1}{294{,}000}} : \underset{\text{Work on Iron.}}{26'' \times 3.1416 \times \frac{26}{3\frac{1}{2}} \times \frac{1}{17{,}000{,}000}} :: .00085 : .0000168;$$

and 8668 : 608 :: 168 : 12. The total actual theoretical pressure is therefore 608 — 12 = 596 pounds per square inch.

	Proportion borne by Iron. Pounds.	Pounds.	Pounds.
We have thus, required to stop velocity of 5 feet	314	— 6	= 308
And double the weight of column	294	— 6	= 288
Total pressure due to the fall			596

This is the simplest form of shock. When a bucket or ram is suddenly stopped in descending, the calculation becomes more complicated from the working-barrel being partly occupied by iron or wood,* which are respectively 60 and 6 times less compressible than water, and from the momentum of the machinery backed by the steam-pressure having also to be met. The calculation, however, of this simple shock

* In mining-pumps only.

makes more intelligible the thickness of iron practically found necessary in pumps, keeping in view that repeated shocks weaken cast iron by a half. It likewise shows that the pressure due to the shock depends upon the velocity of the water when suddenly stopped and the resistance to compression of itself and the resistance to extension of the vessel containing it, and as in an ordinary pressure-gauge the velocity of the water and the resistance to extension of the vessel containing it are both less than in the working-barrel, the pressure indicated must also be less.

Air-vessels will be found useful in lessening shocks in pumps; but it would appear that within the working-barrel they cannot do so to the extent generally supposed. The time taken to compress water sufficiently to stop a 5-feet velocity and raise 314 pounds is only $\frac{1}{3000}$ part of a second. This time must elapse before the water below the air-vessel can begin to flow back, and the extent to which the water above the air-vessel can be relieved of the shock due to its velocity must depend on the quantity of water in the air-vessel and the direction and rate of its motion at the instant, as well as the quantity of air, because the quicker the change of motion and the larger the mass of the water the greater the force required, and an abrupt change of motion requires an infinite force. If there be one foot in length of water between the main column and the vessel, and that at rest at the beginning of the shock, it will attain the same velocity as the descending water when .866 of the full compression due to the shock has been reached; and the pressure will probably not much exceed this proportion thereafter at or above the entrance to the vessel, provided there be sufficient air above the water.

About 1½ feet of air would suffice for a shock capable of producing a pressure of 600 pounds per square inch. No substantial relief would be given if two feet of water intervenes between the main column and the air in the vessel. Water may be much more readily driven in or out of the air-vessel to or from the bucket than can the whole column of

water be moved, because the mass of water to be moved by the same pressure is much less, and probably the air-vessel is as useful in driving the water downwards to fill a vacancy and so permit the fall of the column above as in forming a cushion for the fall.

Pumping Hot Water.—Experiences are not wanting in which the difficulty of pumping hot water has been made clearly manifest, but recorded experiments as to its practical limitations are quite rare. It has been the author's practice to always insist that water having a temperature higher than 120° shall flow into the suction-chamber of the pump; if this be done the matter then becomes simply one of a proper selection of pump and valves.

Mr. H. J. Coles, Inst. C. E., London, made some experiments, the results of which were embodied in a paper presented to the above institution, an abstract of which is here given:

"The depth from which hot water of a given temperature may be pumped can be theoretically deduced from the formula

$$\log. p = A - \frac{B}{t} - \frac{C}{t^2},$$

and the inverse of the above

$$\frac{1}{t} = \sqrt{\left(\frac{A - \log. p}{C} + \frac{B^2}{4C^2}\right)} - \frac{B}{2C}$$

in which p = absolute pressure, t = absolute temperature, and A B C are constants. Both formulas are quoted from page 283 of Rankine's 'Rules and Tables;' or the same results may be more readily obtained from Regnault's Tables given at page 263, 'Ganot's Physics' (third edition).

"Having had, however, frequent inquiries as to what could be done in actual practice by donkey-pumps for feeding boilers from hot-wells, etc., and supposing that possibly sufficient vapor might be evolved from the water at lower temperatures than those ascertained by the above rules to diminish seriously the quantity pumped, the author carried out a series of experiments to obtain actual results.

"The donkey-pump employed was single-acting, having a

ram 3 inches in diameter, with a length of stroke of 7 inches. The pump was elevated to various heights; but the results being so nearly alike, allowing for difference in height and temperature, the table given below for 15 feet may be taken as typical of all. The supply-tank stood on the ground, the water in it being heated by a jet of steam. The suction-pipe was led direct to the valve-box with only one bend, and the delivery-tank was elevated to about the same level as the pump, the water being discharged through a valve loaded to 60 pounds per square inch. A large cock, fitted to the bottom of the delivery-tank, was kept open while the speed of the pump was being regulated, and was shut as soon as the trial commenced. A certain depth of water always existed in the tank while the cock was open; this was carefully gauged and deducted at the end of the trial. The speed of the pump was regulated as nearly as possible to that given in the first column, and on each trial, as soon as the exact number of strokes was completed, the pump was stopped.

"It will be seen that the results agree closely with those given by the above rules, the falling off in the quantity at the higher temperatures being most probably due to the friction of the water in passing through the pipes, valves, etc. It will also be observed that the speed of the pump had to be reduced for the higher temperatures, the speeds stated in the list being found to give the best results.

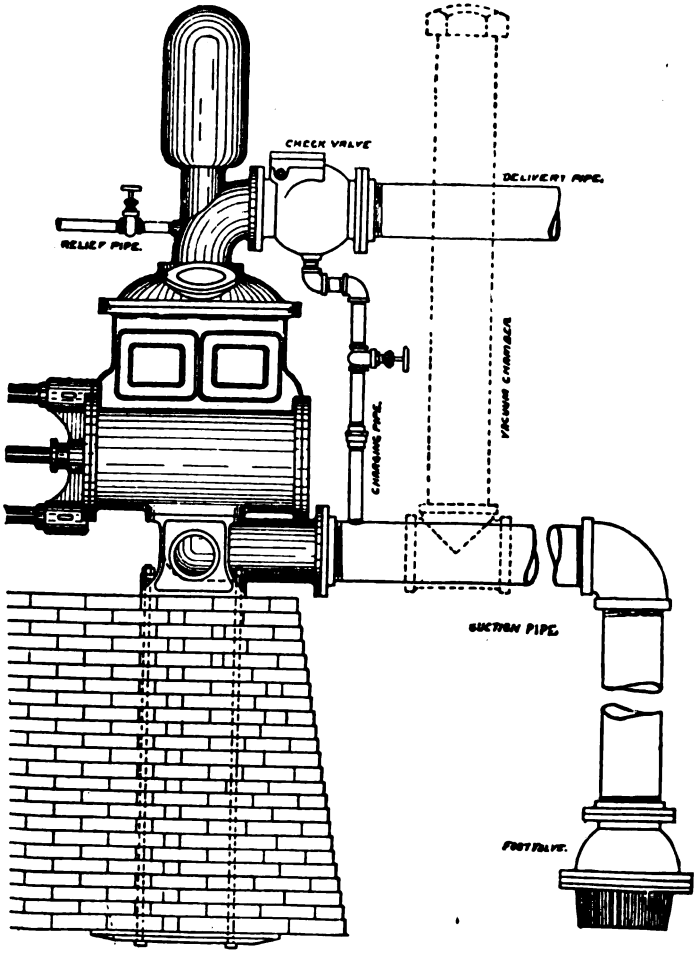
TABLE VII.

RESULTS OF EXPERIMENT NO. 3 WITH THE PUMP 15 FEET ABOVE THE WATER-LEVEL.

Revolutions per Minute.	Temperature, Fahrenheit. Degrees.	Hot Water pumped per Minute. Cubic Inches.
70	70	3430
70	100	3430
70	120	3430
70	140	3430
70	160	3286
60	170	2682
50	180	2180

" Each quantity stated is the mean of several trials. Above 180° Fahrenheit scarcely any water could be pumped. According to Regnault, 185° would be about the limiting temperature at 15 feet."

FIG. 107.



A general arrangement for piping a pump is shown in Fig. 107, subject, of course, to such modifications as are necessary to adapt it to any particular location.

The suction-pipe should be as short and direct as possible, and never less in diameter than the opening leading into the pump. A foot-valve should be used, and if a strainer is not attached to it there should be a strainer-box built around it to keep out floating-matter, fish, etc. The check-valve should bolt directly against the flange of the delivery-elbow.

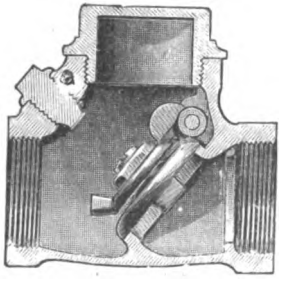
A charging-pipe should lead from the pressure-side of the check-valve into the suction-pipe. By opening the valve shown in the charging-pipe, the water from the reservoir or tank will flow into and fill the suction-pipe and the pump, thus expelling all the air.

A relief-valve and pipe is shown immediately back of the check-valve. This valve should be opened when filling the suction-pipe and pump with water, in order to allow the air to escape. The relief-pipe should lead to a drain or any place where the overflow would not cause inconvenience.

A vacuum-chamber is shown in dotted lines, together with a tee in the suction-pipe. This is not always necessary, but when it is it should be placed near the pump. If the suction-pipe goes into a deep well, and the distance from the well to the pump is short, the delivery-elbow may be changed to a tee, and the vacuum-chamber placed directly over the high lift.

The steam-pipe should be so arranged that the water of condensation, when the pump is not running, will drain back into the boiler. The exhaust may be led to any convenient point for escape into the atmosphere, or it may be used in steam coils for heating.

FIG. 108.



Check-valves for pumps should always be straight way and fitted with hinged or swinging valves, and ought to be attached directly to the delivery-flange of the pump, if practicable. The diameter should equal in size that of the delivery-opening of the pump.

There are several good straight-way valves now furnished the trade in the smaller sizes, so that there is less need than formerly for the employment of spindle check-valves fitted to seats at right angles to the flow. Fig. 108 is a sectional elevation of a Pratt & Cady check-valve which the writer has found to be quite satisfactory. Check-valves for diameters suited to 3-inch wrought-iron pipe and less should always be made of tough brass. For larger sizes a good design is shown in Fig. 109. It will be observed that the valve-face is nearly at right angles to the flow; the position of the fulcrum and the angle of the valve-face insures prompt return to its seat. Such valves are regularly made in all sizes, from 4 inches to 24 inches diameter of opening; larger sizes are made to order, and have usually two or more rectangular hinged valves, rather than one large one. The shell and valve are of cast iron; the valve is faced with leather and held in place by a brass ring or plate underneath; throughgoing rivets securely fasten all three together, a detail not shown in the drawing.

FIG. 109.

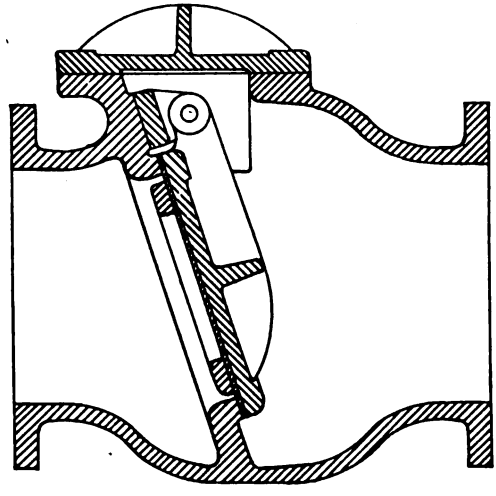


TABLE VIII.

PRESSURE OF WATER.

The Pressure of Water in Pounds per Square Inch for every Foot in Height to 300 Feet; and then by Intervals, to 1000 Feet Head.

Feet Head.	Pressure per Square Inch.	Feet Head	Pressure per Square Inch.	Feet Head.	Pressure per Square Inch.	Feet Head.	Pressure per Square Inch.	Feet Head.	Pressure per Square Inch.
1	0.43	44	19.05	87	37.68	130	56.31	173	74.94
2	0.86	45	19.49	88	38.12	131	56.74	174	75.37
3	1.30	46	19.92	89	38.55	132	57.18	175	75.80
4	1.73	47	20.35	90	38.98	133	57.61	176	76.23
5	2.16	48	20.79	91	39.42	134	58.04	177	76.67
6	2.59	49	21.22	92	39.85	135	58.48	178	77.10
7	3.03	50	21.65	93	40.28	136	58.91	179	77.53
8	3.46	51	22.09	94	40.72	137	59.34	180	77.97
9	3 89	52	22.52	95	41.15	138	59.77	181	78.40
10	4.33	53	22.95	96	41.58	139	60.21	182	78.84
11	4.76	54	23.39	97	42.01	140	60.64	183	79.27
12	5.20	55	23.82	98	42.45	141	61.07	184	79.70
13	5.63	56	24.26	99	42.88	142	61.51	185	80.14
14	6.06	57	24.69	100	43.31	143	61.94	186	80.57
15	6.49	58	25.12	101	43.75	144	62.37	187	81.00
16	6.93	59	25.55	102	44.18	145	62.81	188	81.43
17	7.36	60	25.99	103	44.61	146	63.24	189	81.87
18	7.79	61	26.42	104	45 05	147	63.67	190	82.30
19	8.22	62	26.85	105	45.48	148	64.10	191	82.73
20	8.66	63	27.29	106	45.91	149	64.54	192	83.17
21	9.09	64	27.72	107	46.34	150	64.97	193	83.60
22	9.53	65	28.15	108	46.78	151	65.40	194	84.03
23	9.96	66	28.58	109	47.21	152	65.84	195	84.47
24	10.39	67	29.02	110	47.64	153	66.27	196	84.90
25	10.82	68	29.45	111	48.08	154	66 70	197	85.33
26	11.26	69	29.88	112	48.51	155	67.14	198	85.76
27	11.69	70	30.32	113	48.94	156	67.57	199	86.20
28	12 12	71	30.75	114	49.38	157	68.00	200	86.63
29	12 55	72	31.18	115	49.81	158	68.43	201	87.07
30	12.99	73	31.62	116	50.24	159	68.87	202	87.50
31	13.42	74	32.05	117	50.68	160	69 31	203	87.93
32	13.86	75	32.48	118	51.11	161	69.74	204	88.36
33	14.29	76	32.92	119	51.54	162	70.17	205	88.80
34	14.72	77	33.35	120	51.98	163	70 61	206	89.23
35	15.16	78	33.78	121	52.41	164	71.04	207	89.66
36	15.59	79	34.21	122	52.84	165	71.47	208	90.10
37	16 02	80	34.65	123	53.28	166	71.91	209	90.53
38	16.45	81	35.08	124	53.71	167	72.34	210	90.96
39	16.89	82	35.52	125	54.15	168	72.77	211	91.39
40	17.32	83	35.95	126	54.58	169	73.20	212	91.83
41	17.75	84	36.39	127	55.01	170	73.64	213	92.26
42	18.19	85	36.82	128	55.44	171	74.07	214	92 69
43	18 62	86	37.25	129	55.88	172	74.50	215	93.13

TABLE VIII.—(*Continued.*)

Feet Head	Pressure per Square Inch.	Feet Head.	Pressure per Square Inch.	Feet Head.	Pressure per Square Inch.	Feet Head.	Pressure per Square Inch.	Feet Head.	Pressure per Square Inch.
216	93.56	237	102.60	258	111.76	279	120.85	300	129.95
217	93.99	238	103.09	259	112.19	280	121.29	310	134.28
218	94.43	239	103.53	260	112.62	281	121.72	320	138.62
219	94.86	240	103.96	261	113.06	282	122.15	330	142.95
220	95.30	241	104.39	262	113.49	283	122.59	340	147.28
221	95.73	242	104.83	263	113.92	284	123.02	350	151.61
222	96.16	243	105.26	264	114.36	285	123.45	360	155.94
223	96.60	244	105.69	265	114.79	286	123.89	370	160.27
224	97.03	245	106.13	266	115.22	287	124.32	380	164.61
225	97.46	246	106.56	267	115.66	288	124.75	390	168.94
226	97.90	247	106.99	268	116.09	289	125.18	400	173.27
227	98.33	248	107.43	269	116.52	290	125.62	500	216.58
228	98.76	249	107.86	270	116.96	291	126.05	600	259.90
229	99.20	250	108.29	271	117.39	292	126.48	700	303.22
230	99.63	251	108.73	272	117.82	293	126.92	800	346.54
231	100.06	252	109.16	273	118.26	294	127.35	900	389.86
232	100.49	253	109.59	274	118.69	295	127.78	1000	433.18
233	100.93	254	110.03	275	119.12	296	128.22		
234	101.36	255	110.46	276	119.56	297	128.65		
235	101.79	256	110.89	277	119.99	298	129.08		
236	100.23	257	111.32	278	120.42	299	129.51		

CHAPTER VII.

WATER-END DESIGN.

THIS subject has already been touched upon, and especially in the chapter on valves, as the latter detail is so intimately connected with water-end design that it is not easy to separate them. The illustrations given in this chapter are so nearly self-explanatory that little descriptive matter will be necessary. The writer regrets that the small size of the engravings necessary to their insertion in a printed page of the dimensions selected for this publication would not permit of both scale and dimensioned drawings to be given, but the illustrations themselves represent designs now in successful use, so that the mere fact of such presentation in convenient form will probably not fail to be useful to engineers and others not interested in the design and manufacture of pumping machinery, but who are desirous of knowing the interior arrangement of the pumps now offered by builders to the general public.

Length of Stroke and Piston-Speed.—In ordinary short-stroke pumps the capacity of the water-end is limited not so much to the piston-speed in feet per minute as by the number of times a valve can safely and noiselessly open and close in a given time. To assume 100 feet per minute as an ordinary speed for pumps has been a time-honored practice; it is obvious, however, that for short strokes it imposes an injurious rate of speed, to which a pump should not, in regular service, be subjected. For example, a pump having

3-inches stroke must make 400 strokes per minute.

4	"	"	"	"	300	"	"	"
5	"	"	"	"	240	"	"	"
6	"	"	"	"	200	"	"	"
7	"	"	"	"	171 +	"	"	"
8	"	"	"	"	150	"	"	"
10	"	"	"	"	120	"	"	"
12	"	"	"	"	100	"	"	"

As the above list of strokes represents lengths commonly in use, it needs no argument to show the impracticability of the 100-feet basis of comparison; for pumps having a stroke of six inches and less, the number of strokes as given above is too great for continuous service.

TABLE IX.

SPEED AND CAPACITY OF PUMPS.

A piston-speed of 100 feet per minute is considered an ordinary speed for direct-acting pumps. But in boiler-feeding, pumping under heavy pressure, or moving hot liquids, a slower speed is advisable. In fire-pumps, on the contrary, where a high velocity and large volume are imperative, the speed may exceed 200 feet per minute, if the valve area of the pump is sufficiently large.

THEORETICAL CAPACITY OF 100 FEET SPEED OF PISTON OR PLUNGER PER MINUTE.

Diameter of Pump or Plunger in Inches.	Gallons discharged per Minute.	Diameter of Pump or Plunger in Inches.	Gallons discharged per Minute.	Diameter of Pump or Plunger in Inches.	Gallons discharged per Minute.	Diameter of Pump or Plunger in Inches.	Gallons discharged per Minute.
1	4.08	3	36 75	6	147	14	800
1 1/8	5.16	3 1/4	43.13	6 1/2	172	15	917
1 1/4	6.38	3 1/2	50.02	7	200	16	1044
1 3/8	7.71	3 3/4	57.42	7 1/2	229	18	1321
1 1/2	9.18	4	65.34	8	261	20	1631
1 5/8	10.78	4 1/4	73.76	8 1/2	295	22	1974
1 3/4	12.58	4 1/2	82.7	9	330	24	2350
1 7/8	14.35	4 3/4	92.14	9 1/2	368	26	2757
2	16 33	5	102.0	10	408	28	3270
2 1/4	20.67	5 1/4	112.0	10 1/2	450	30	3670
2 1/2	25.52	5 1/2	123.0	11	494	32	4176
2 3/4	30.88	5 3/4	135.0	12	587	34	4715

This is the theoretical performance; practically the strokes will slightly exceed that number for the quantity stated. In a duplex pump the number of gallons delivered per minute is found by multiplying the displacement of one plunger by twice the number of strokes.

TABLE X.

SHOWING THE NUMBER OF STROKES REQUIRED TO ATTAIN A PISTON-SPEED FROM 50 TO 125 FEET PER MINUTE FOR PUMPS HAVING STROKES FROM 3 TO 18 INCHES IN LENGTH.

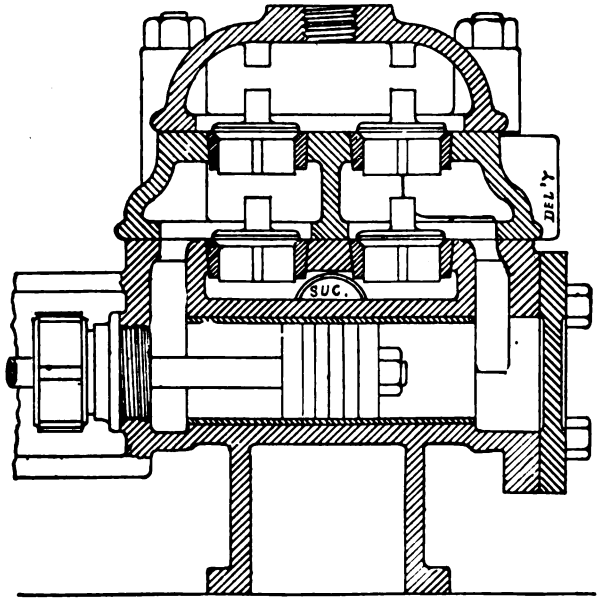
Speed of Piston in Feet per Minute.	Length of Stroke in Inches.									
	3	4	5	6	7	8	10	12	15	18
	Number of Strokes per Minute.									
50	200	150	120	100	86	75.0	60	50	40	33
55	220	165	132	110	94	82.5	66	55	44	37
60	240	180	144	120	103	90.0	72	60	48	40
65	260	195	156	130	111	97.5	78	65	52	43
70	280	210	168	140	120	105.0	84	70	56	47
75	300	225	180	150	128	112.5	90	75	60	50
80	320	240	192	160	137	120.0	96	80	64	53
85	340	255	204	170	146	127.5	102	85	68	57
90	360	270	216	180	154	135.0	108	90	72	60
95	380	285	228	190	163	142.5	114	95	76	63
100	400	300	240	200	171	150.0	120	100	80	67
105	420	315	252	210	180	157.5	126	105	84	70
110	440	330	264	220	188	165.0	132	110	88	73
115	460	345	276	230	197	172.5	138	115	92	77
120	480	360	288	240	206	180.0	144	120	96	80
125	500	375	300	250	214	187.5	150	125	100	83

Note.—To find the number of plunger displacements in a duplex pump multiply the number of strokes as given above by 2.

Piston Water-End with Wing-Valves.—Fig. 110 is a sectional elevation of a duplex pump for feeding steam-boilers, or for any other service where a pressure not exceeding 150 pounds per square inch may be required; it is of the piston pattern, and lined with a drawn brass tube, carefully fitted and then forced into the bored casting, after which the ends of the tube are expanded by calking or riveting over a slight depression included in the casting. The holes for the valve-seats may be bored with a straight taper of say 1 inch to the foot, as shown in the engraving, or a taper tap may be used and the valve-seat screwed in. This water-end represents a pump of 2 inches bore by 4 inches stroke, and for this size it has been the practice to drive the valve-seats in on a taper. This drawing shows a joint immediately above the water-cylinder, and

another joint immediately above the top of the delivery-valve seats, the delivery- or force-chamber completing the main parts. The holding-down bolts extend through from the top of the lugs included in the force-chamber casting, and screw into

FIG. 110.



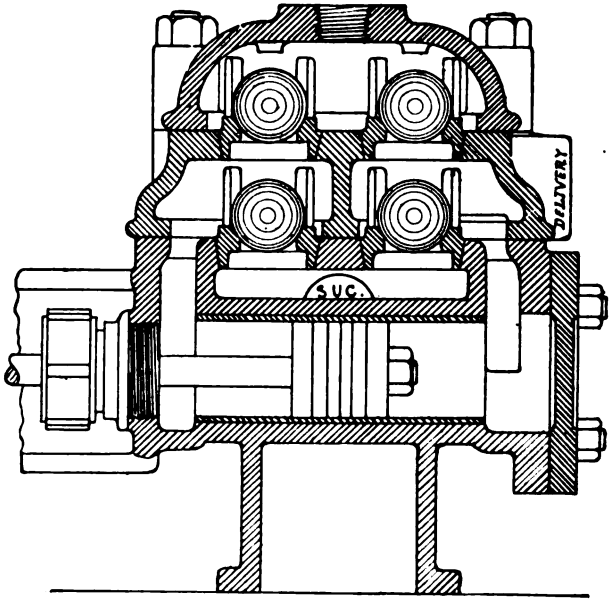
tapped holes in the water-cylinders below. The piston may be for fibrous packing, or fitted with metal rings. The valves are of gun-metal, as are also the seats. This drawing is shown with wing-valves with mitre joints.

Piston Water-End with Ball-Valves.—Fig. 111 is the same water-end as described above, except that ball-valves and seats are fitted for the handling of thick stuffs like molasses, etc.

Plunger-Pump.—A sectional elevation of a water-end having a plunger and ring is shown in Fig. 112. The ring

slides into a bored cavity in the water-cylinder and is held in place by bolts, one of which is shown in the drawing. The plunger slides through the ring, and is not provided with any means of adjustment for wear. The drawing is from a pump having plungers 4 inches in diameter by 6 inches stroke. The

FIG. 111.

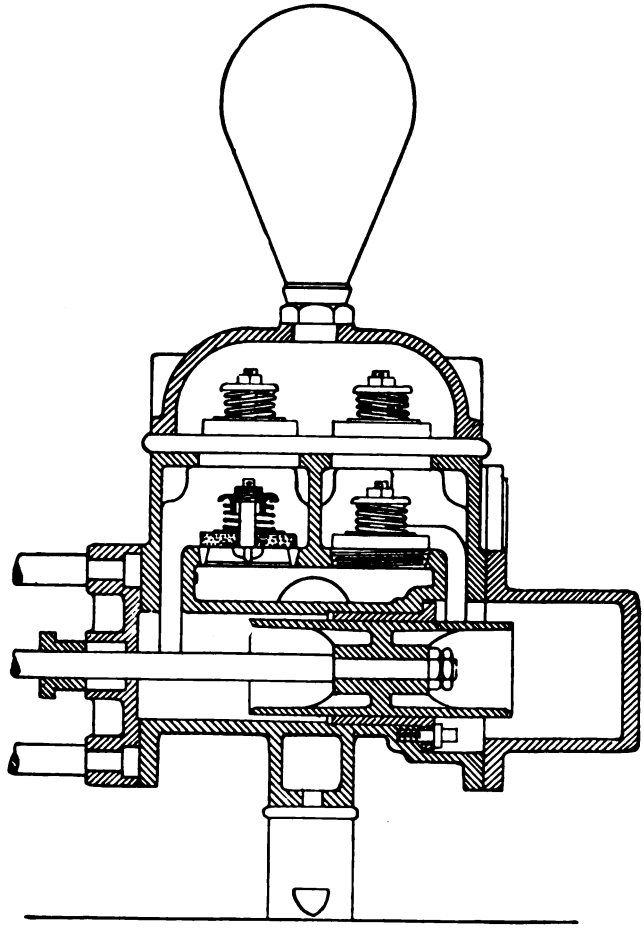


valve-seats are screwed in; the valves are of india-rubber for cold water, and may be fitted with metal valves or vulcanite composition, if so desired, for hot water. The delivery-valve seats are screwed into a valve-plate held between the water-end casting and the force-chamber.

A piston water-end with a removable lining is shown in Fig. 113. The water-cylinder is bored at the three or more points of support, and the lining is turned to fit; the support or rib at the rear end is machine-faced, as is also the lining, so as to make true surfaces for bolting together. Linings

should be made of gun-metal, and must be sound castings, free from imperfections and accurately bored. By a proper spacing of the drilling the lining can be arranged for turning

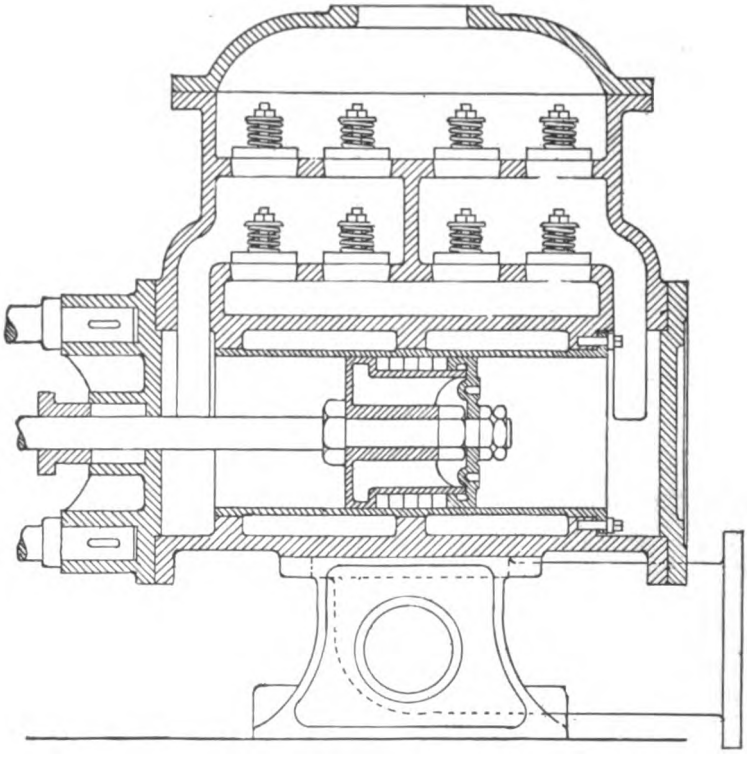
FIG. 112.



around in position so as to present a new wearing surface on the bottom, should it ever become necessary through the abrasive action of gritty water passing through the pump.

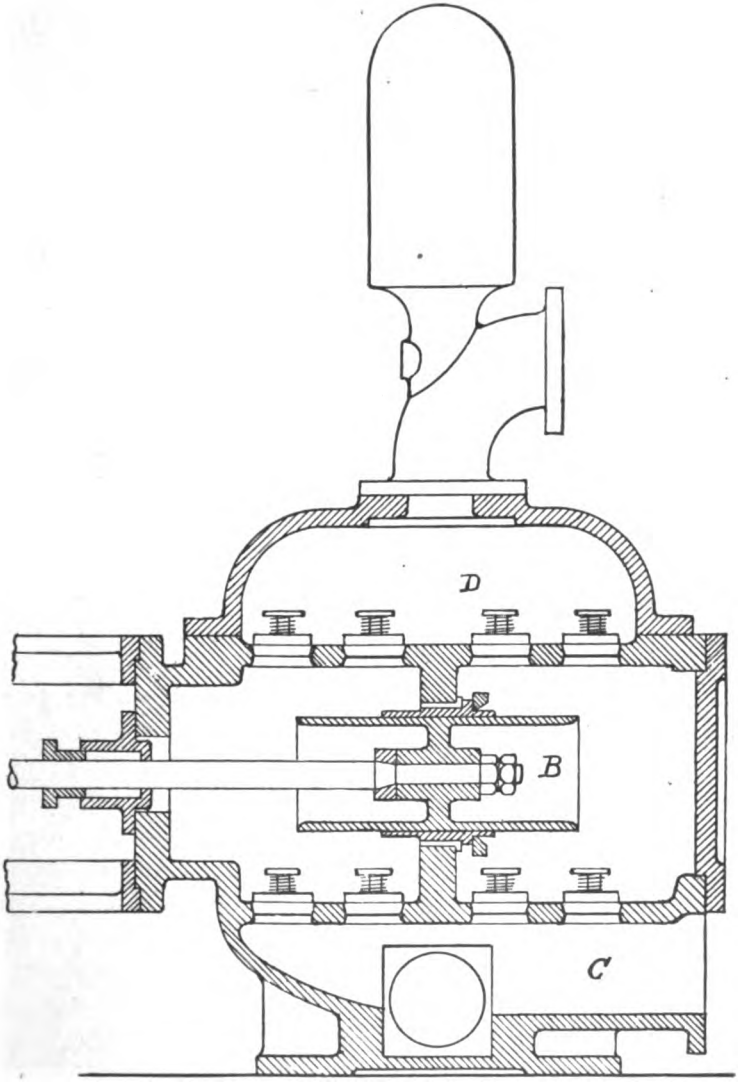
The suction- and delivery-valves are both above the piston, so that once the pump is charged with water it will always remain so. The valve-seats are intended to be screwed in. This drawing partially represents a duplex water-end having 7 inches bore by 12 inches stroke.

FIG. 113.



Worthington Plunger-Pump.—The sectional elevation of a Worthington plunger-pump, shown in Fig. 114, is thus described in their catalogue: The double-acting plunger shown at *B* works through a deep metallic packing-ring, bored to an accurate fit, being neither elastic nor adjustable. Both the ring and the plunger can be quickly taken out, and either refitted or exchanged for new ones at small cost, and

FIG. 114.



if it be desired at any time to change the proportions between the steam-pistons and pumps, a plunger of somewhat larger size, or decreased to any smaller diameter, can be

readily substituted. As exact proportions between the power and work are always desirable, if not necessary, this is a very important advantage.

This system of renewal of the working parts has proved by long experience to be the least expensive and most satisfactory for ordinary work. The plunger is located some inches above the suction-valves to form a subsiding-chamber, into which any foreign substances may fall below the wearing-surfaces. This enables it to work longer without renewal than the usual form of piston-pump, especially in water containing grit or other solid material. The water enters the pump through the suction-chamber *C*, through the suction-valves, then passes partly around and partly by the end of the plunger, through the force-valves, nearly in a straight course, into the delivery-chamber *D*, thus traversing in a very direct and ample water-way. The bottom and top plates furnish a large area for the accommodation of the valves. These consist of several small disks of rubber, or other suitable material, easy to examine and inexpensive to replace.

The drawing shows the usual central partition separating the two ends of the pump. It is the practice to bore out this central partition and then insert a permanent collared brass bushing, into which is fitted the plunger-ring, also provided with a collar, one side of which abuts against the bushing, and the other side adapted to receive a cast-iron ring for holding in place, this latter ring being securely held by bolts not shown in the drawing.

The accumulated facts of a large and successful experience with plungers simply fitted into a bored ring, as compared with packed plungers, so clearly set forth by the late Mr. Worthington, cannot fail to interest the reader. His close study of the peculiarities of action of his duplex pump led him very early in his practice to adopt a metal packing-ring, without elasticity or provision for adjustment, a detail which, after forty years of continuous trial, is the only form of packing ordinarily recommended by his successors, because nothing better has been found to take its place. Let it be noticed

that engines of the duplex type have little or no momentum to help out the stroke; they therefore will not allow of any inequalities or tight places in the packing. A cylinder or a plunger always wears fastest in the middle of the stroke; if the packing were adjusted to fit the middle, it would bind enough at the ends to embarrass the motion. He detected a tendency at times in elastic pump-packings to collapse upon a plunger or expand against a cylinder with such force as to produce much unnecessary friction.

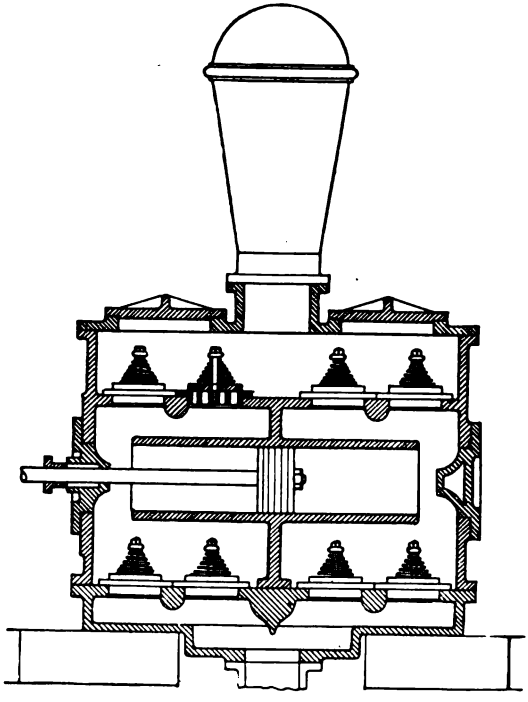
The wear of the plungers in the non-adjustable metallic rings is not as great as one might be led to suppose, and for reasons that will appear obvious on examination. The rings are made deep, and thus afford ample bearing surfaces. The water-ways surrounding the plungers constitute subsiding-chambers into which hurtful material can settle away from the moving parts. (See Fig. 114.) The protrusion into the forcing side of the pump of the plunger while in motion tends to carry away from the ring any such material that would otherwise be forced under it. The thin film of fluid that may enter the space between the ring and the plunger is only sufficient for proper lubrication of the parts, and reduces their frictional resistance to a minimum. As the pressures are reversed when the pump changes its stroke, little or none of this water leaks past the ring.

In such cases, however, as seem, by reason of unusually gritty water or of excessive pressure, to demand adjustable packing, any one of the several designs shown in this chapter will give satisfaction, if the details are properly worked out. All of these packed plunger water-ends, except one design, are arranged with exterior stuffing-boxes, so that the packing is readily inserted and adjusted to compensate for wear. The stuffing-box glands act as sufficient guides or bearings for the plungers. All leakage in the water-cylinders is in this arrangement prevented, as the working parts are displayed to the engineer at all times. This form of water-end is largely used, especially in the Western States, for water-works supply; it is the type almost always selected for mines and for

iron- and steel-works. Great care must be taken in the design and construction of pumps of this type to reduce as much as practicable the large loss inseparable from packed plungers of all forms. This loss, even under the best conditions of construction and careful packing, consumes a large percentage of the power applied, especially when the pump is running against heavy pressures. In order to diminish the friction as much as possible, it is recommended that the stuffing-box glands be screwed up no tighter than is actually necessary to prevent leakage.

A piston water-end by the Erste Brünner Maschinen-Fabriks, Brünn, having some of the characteristics of the Worthington water-end, is shown in Fig. 115. By reason of the working-barrel being included in the main casting of the water-end, it is less difficult to construct if the latter is made in two pieces, as shown in the drawing; it is true an additional joint is required, but the facility with which the work can be done in the machine-shop will probably not increase the cost of the water-end.

FIG. 115.



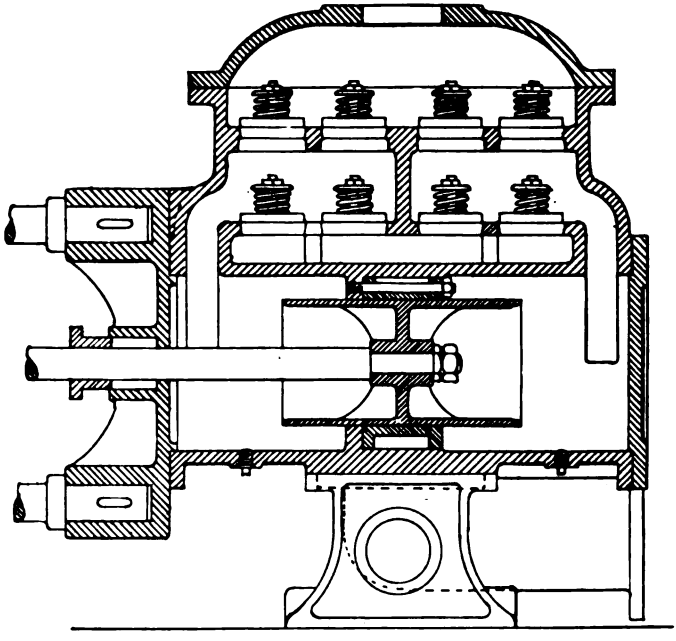
This illustration is from one of a pair of pumps placed side by side, each pump being driven by an automatic cut-off engine, both of which are coupled to the same shaft, and have one fly-wheel in common, but each engine may with its own pump be operated singly if so required. The pumps are 11.9 inches diameter by 43.55 inches stroke. Each end of each pump has two suction- and two discharge-valves as shown; the valves are 10.7 inches diameter, are of india-rubber, and have wrought-iron back-plates held down by conical spiral springs. The valves are fitted with brass bushings to reduce the wear in the central hole. The valve-gratings are of cast iron. The air-chamber seems very small for a pump as large as the one now under consideration, but it is supplemented by a wrought-iron "wind-chest," about 36 inches diameter by 10 feet in height, situated to the rear of and central to the two water-ends. This vessel is divided into two chambers by a wrought-iron diaphragm, one of which acts as an air-vessel for the suction, the other half acting for the delivery.

The plunger-pump shown in Fig. 116 is from designs by the author, who recognizes fully the value of a larger number of small valves rather than a less number of large valves, provided the latter are of a diameter exceeding 4½ inches. The plunger-ring is supported at each end, in a bored recess in the water-cylinder, and is securely bolted to a flange which is cast in and forms a part of the main cylinder casting, insuring not only a perfect alignment, but the utmost rigidity. The plungers and rings are easily removed from the pump whenever desired, either for examination or repairs.

The discharge- and suction-valves are above the plungers; the pump can never, therefore, lose its charge. This feature is believed to be of value to the user in securing prompt action in emergency and avoiding the danger of breakage that is incurred with pumps running dry through leakage past the suction-valves, when the latter are located below the plungers. Hand-hole openings permit an easy examination of both the

suction- and discharge-valves, and without drawing off the water from the main cylinders of the pump.

FIG. 116.

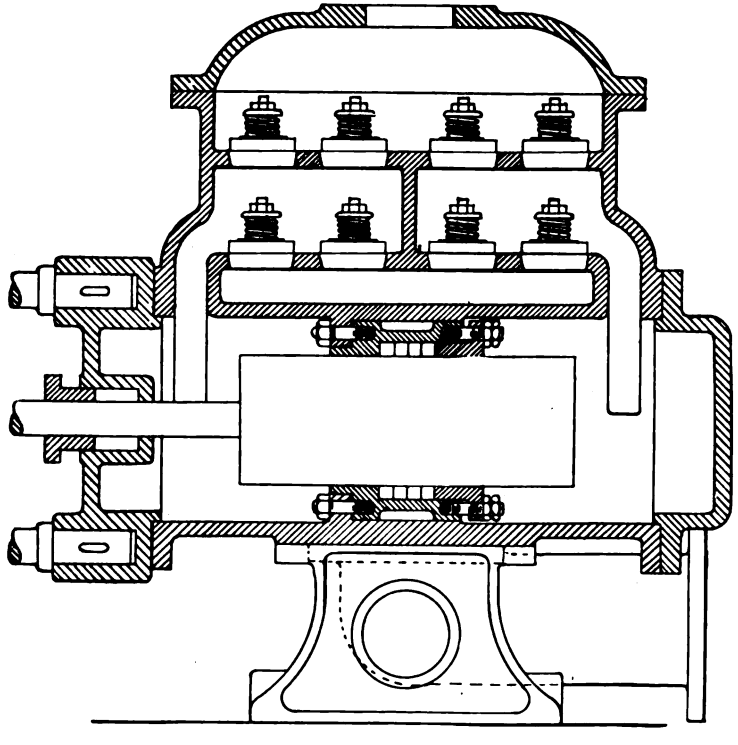


The inside-packed plunger-pump illustrated in Fig. 117 is a modified form of the water-end just described; it preserves the distinctive features of the above pump, the addition of a stuffing-box and gland, adapted for packing the plunger with a flexible or a fibrous packing, constituting the only change. This pump is chiefly desirable in localities where the water-supply is muddy or contains gritty matter.

This form of pump is to be preferred over a piston-pump for gritty water, inasmuch as the cost of a new plunger is very much less than that of a new lining. There is the further advantage in the fact that slight reductions can be made in the diameter of the plunger by turning in a lathe to remove the

scoring incident to the service in which it may be employed, the difference in diameter being made good by the use of a thicker packing.

FIG. 117.

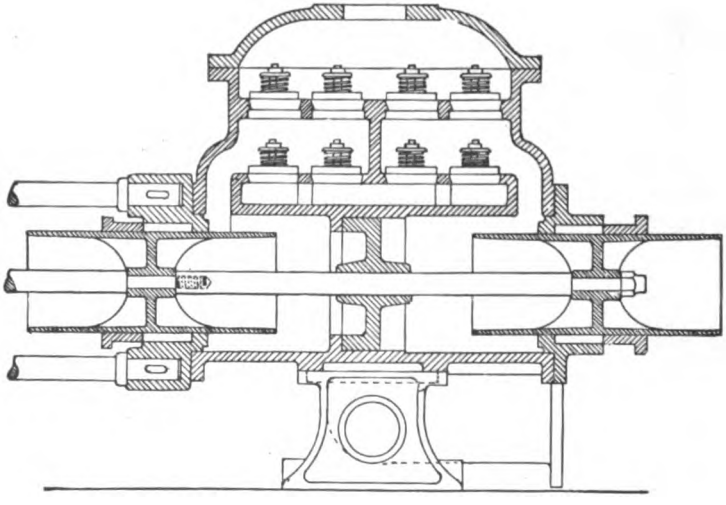


The outside-packed plunger-pump shown in sectional elevation in Fig. 118 is still another modification of the plunger-pump (Fig. 116). Instead of the plunger-ring as there shown, a central diaphragm, bored to receive the rod connecting the two plungers, is secured by bolting in the bored recess contained in the water-cylinder. This diaphragm is not fitted with a stuffing-box, nor is it the common practice to include a brass bushing, the cast iron forming of itself a suitable material for resisting the very slight wear of the rod passing

through it. Instead of the ordinary heads, those fitted with a stuffing-box and gland, as shown in the drawing, are substituted, through which the plungers slide. The plunger-rod is connected by a screw-joint immediately inside of the inside plunger. The other details are in no respect different from those already described.

Outside-packed plunger-pumps are a more recent invention than the packed piston, their first employment being, in all probability, an expedient by which to obtain a weighted, ver-

FIG. 118.

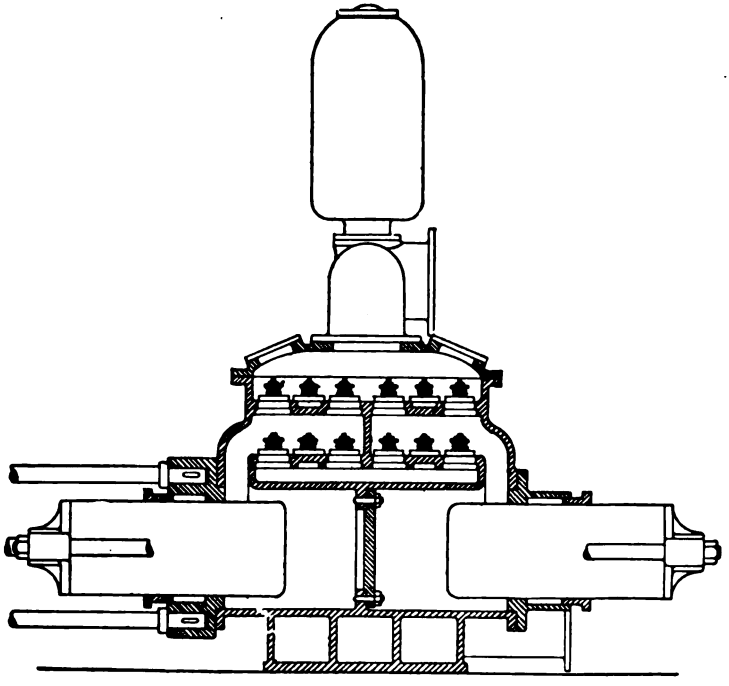


tical, single-acting, elongated piston. This was made to pass through a stuffing-box having leather packing not unlike that used in hydraulic presses at the present time. As this device did not require the boring of the water-cylinder, as must be done in all piston-pumps, together with the fact that the operation f turning is a much simpler and cheaper method of manufacture, it led to the early adoption of the packed plunger on purely commercial principles. Aside from this, there is a certain satisfaction in seeing a plunger pass out and

in a water-end, without leakage, through an adjustable stuffing-box; and this, no doubt, has had its effect in giving direction favorable to this detail in pump design.

The outside-packed plunger-pump with parallel rods, as shown in Fig. 119, is a type of pump in very general favor among furnace, rolling-mill, and steel-works managers. These pumps usually have strokes of 18 to

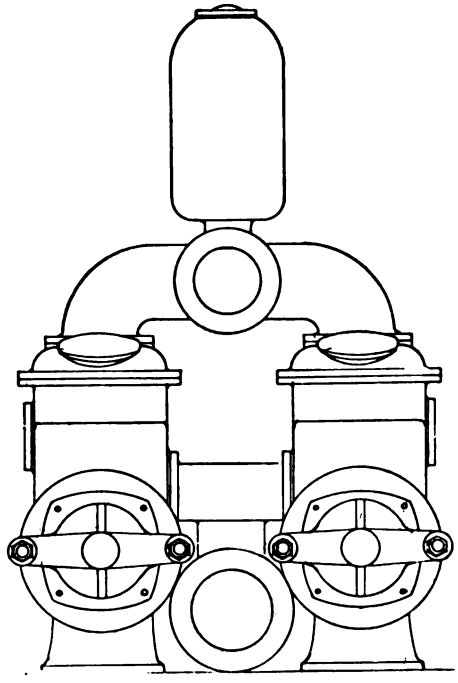
FIG. 119.



24 inches, and are seldom made with plungers less than 12 inches in diameter. The sectional elevation shows clearly the internal arrangement. An end elevation of this pump is shown in Fig. 120. The suction-pipe distributes its flow of water to the two sides of the pump into the space below the

lower or suction-valves. The delivery is central between the two pumps. The air-chamber is located immediately above the delivery-opening. These pumps are generally used at pressures seldom exceeding 75 pounds per square inch.

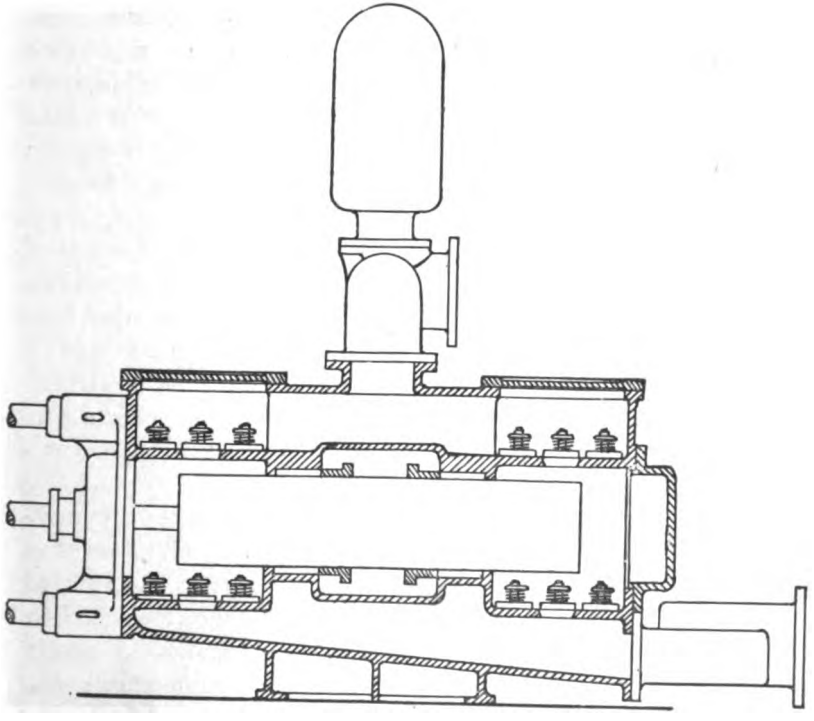
FIG. 120.



An outside centrally-packed plunger-pump is shown in sectional elevation in Fig. 121 and in end elevation in Fig. 122. For ordinary service, pumps of this design range in size from small boiler feed-pumps to sizes suitable for water-works. A modification of this design is employed in mines and other heavier pressures than occur in domestic water-supply. This design includes two separate or single water-cylinders placed side by side, and adapted for duplex steam-cylinders and valve-motion. The suction-opening is at the rear end of each water-cylinder, the two openings being connected by a distributing-pipe, as shown in the drawing. In this design the suction-valves are placed below the plunger, and the delivery-valves above. A delivery-pipe, connecting the two water-ends with air-chamber above, is shown with the flanged opening leading to the rear of the pumps, but it is evident that, by a suitable arrangement of the pattern, it can lead over either side if so desired.

Differential Plunger-Pump.—The combined piston- and plunger-pump, commonly known as the differential plunger-pump, is shown in one of its various forms, in sectional elevation, in Fig. 26. The description of the working of a bucket- and plunger-pump, as given on page 34, is applicable to this design and need not be duplicated. The pump there

FIG. 121.

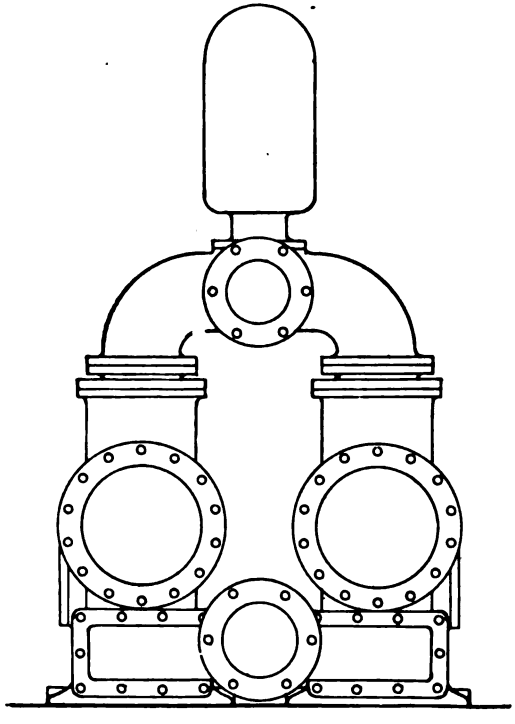


illustrated is of 12 inches stroke, and as it was designed for use as a sinking-pump in mining operations in which acid water is almost certain to be encountered, the valves were all included in the bottom section of the pump. This piece may be made wholly of gun-metal if thought desirable; in any

event it is easily detached from the pump if any repairs or renewals are rendered necessary through the chemical action of acidulated water.

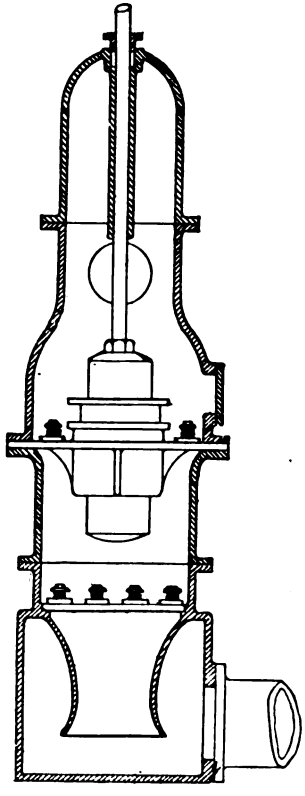
Vertical Plunger-Pump.—A sectional elevation of a vertical plunger-pump, with internal stuffing-box, is shown in Fig. 123, and presents several peculiarities of design which may be considered good: its compactness, in which a suction air-chamber, suction-pipe, and suction-valve plate are included in a single unit, as it were; then the distance-piece to the delivery-valve plate through which the plunger works; the casting containing the delivery-opening, on top of which is placed the air-chamber, makes a good arrangement of detail for some certain positions in which height is available rather than floor-space. The pump is single-acting. The delivery-valve plate shows but two valves, but it will be understood that other valves extend around the whole circumference on properly-spaced centres.

FIG. 122.



A duplex piston-pump with metal clack-valves is shown in longitudinal sectional elevation in Fig. 124 and in cross-section in Fig. 125. This pump was designed by the writer for handling thick stuff, such as mash in brewing operations. The main castings are of iron; the piston, piston-rod, and cylinder-linings are of gun-metal; the valve-seats and valves are hinged together, and are easily removable from the pump through the hand-hole plates opposite each. The lower hand-hole plates may be removed without disturbing the tension of the spring by a simple device secured by hooked bolts shown in the drawing.

FIG. 123.

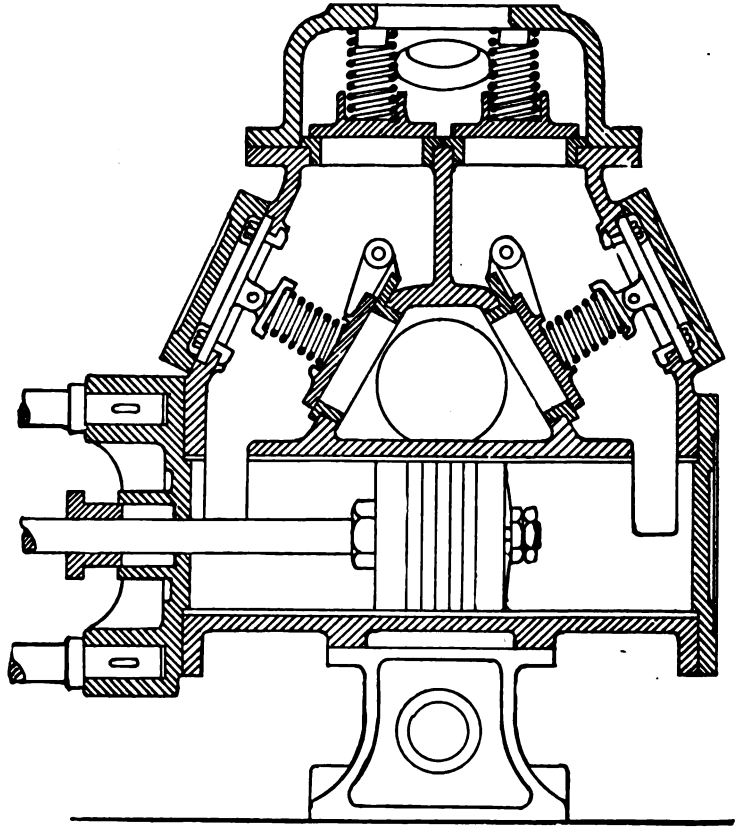


Piston-Pump with Wing-Valves on the Side. — The cross-sectional elevation of a piston-pump with valves on the side, as shown in Fig. 126, is of English design, and represents a type of water-end not often met with in this country. Aside from what the writer believes to be a defective method of guiding the valves by spindles projecting from the top of each, the design is a good one, but is not so compact and symmetrical as the single water-ends furnished by Knowles and other well-known builders.

A by-pass is an attachment to the water-end of a double-acting pump, consisting of a pipe and gate-valve, as shown in the sketch, Fig. 127, the object of which is to make a direct

water connection from one end of the water-cylinder to the other. This attachment is useful in several respects, for example, in a compound pumping engine in which the high-pressure cylinder may not be large enough to start the load from a state of rest, no provision having been made for directly

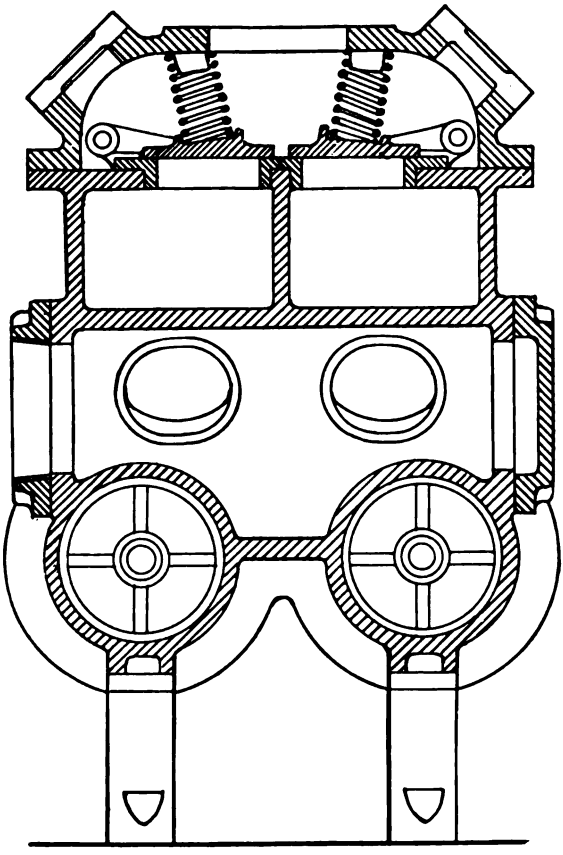
FIG 124.



admitting steam of boiler-pressure in the low-pressure steam-chest. If, now, the by-pass valve be opened, the effect will be to immediately reduce the resistance to the movement of the plunger by allowing the water to flow through from the

pressure side into the one which has no pressure, and thereby enable the engine to complete a sufficient number of strokes to bring the low-pressure cylinder into service; as soon as

FIG. 125.



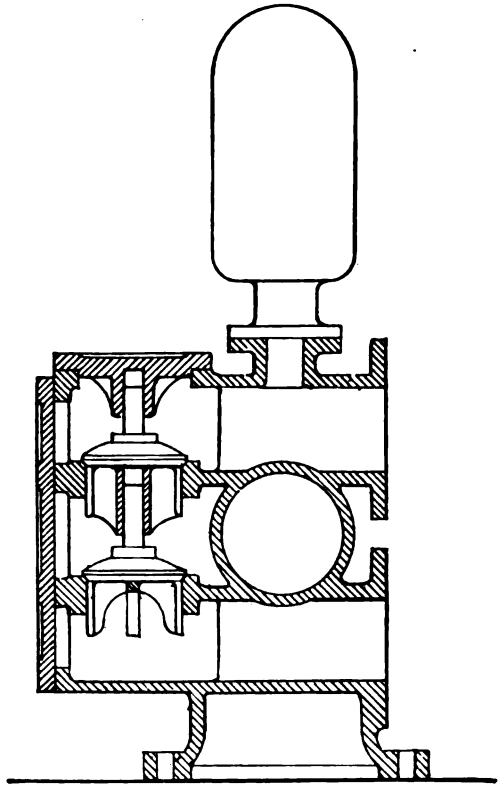
this is accomplished, the by-pass valve should then be closed and the engine allowed to take up its full load.

In crank and fly-wheel engines, especially those operating a direct service, there occur times when the speed of the pump cannot be brought sufficiently low to enable the engine to

properly pass the centres, but if the by-pass valve be opened, the speed of the engine may be increased somewhat without increasing the small quantity of water to be delivered.

It will be understood that a by-pass is simply a leak under the control of the engineer in charge; the size of the opening is not subject to any fixed ratio to plunger area, but it ought to be liberal, and the writer suggests 1½ inches for a 12-inch water-cylinder, and 2½ inches for one 24 inches diameter. A gate-valve only should be used in a by-pass unless it is combined with a charging-pipe, in which case three valves will be required, as shown in the sketch, Fig. 127 A. The valve leading from the supply should be a gate-valve, and the two end ones may be angle-valves.

FIG. 126.

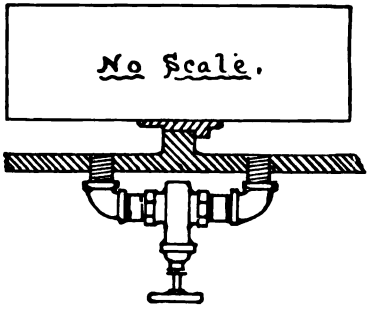


That a by-pass is wasteful of steam is admitted, but its use is entirely one of expediency, and is only resorted to in cases of emergency; so far, then, its application is justified, and should be included in all crank and fly-wheel water-ends intended for direct service. Direct-acting duplex pumps do not require

a by-pass, as they can be run at any rate of speed, however slow.

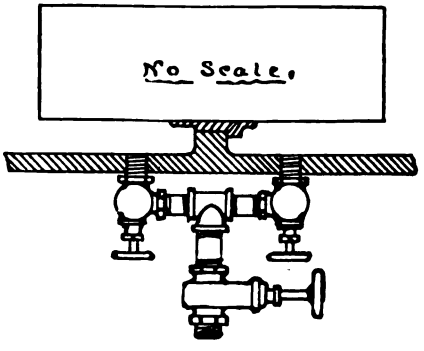
A water-end with bored valve-seats, as shown in Fig. 128, has been adopted by the Woodward Steam Pump Company. The arrows indicate the flow of the water through the pump when in action, on its forward stroke. A valve, the two heads covering the end openings, the bolt for securing in place, are all so clearly shown in Fig. 128 A as to need no further description.

FIG. 127.



The loss of efficiency in pumps under certain conditions, in which the delivery is much less than was to have been expected from the calculated displacement of the plunger and its rate of speed, is at best a difficult thing to trace. So also the loss of power in pumping, in which a large percentage of the energy of the steam-end cannot be accounted for in the delivery of water under a given head. Some experiments witnessed by the writer in a mine of moderate depth (about 450 feet) showed by the indicator diagrams that nearly twenty-five per cent. more power was developed in the steam-end than could be accounted for by the indicator diagrams of the water-end. The pump was direct-acting, and one of many of the same type, this especial one having been selected in

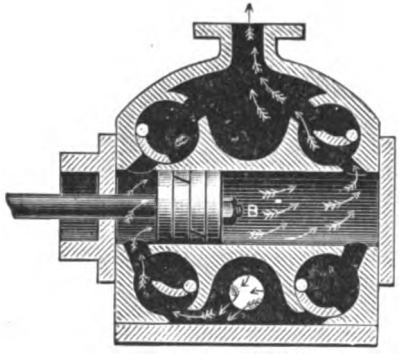
FIG. 127 A.



order to get a high average performance. In this case the suction-pipe was not of excessive length, was of large diameter, and all the conditions apparently favorable to good performance.

Losses occur in pumping if the suction-valves are of such size or shape as not to readily admit the water into the pump-barrel so as to completely fill it by the time the plunger begins its return stroke. The plunger receives its first shock when the body of water is encountered and made to fill the space instantly under full pressure, and its second shock almost immediately thereafter, when the delivery-valves must be raised from their seats. How much power is lost by these two faulty actions is not known, but it is considerable.

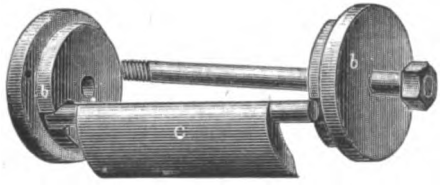
FIG. 128.



The frictional resistance retarding the water is subject to three laws,—quite the reverse of friction between rigid bodies:

1. It is proportioned to the amount of surface in contact.
2. It is independent of the pressure.
3. It is proportional to the square of the velocity.

FIG. 128 A.



From the first of these laws it will be seen that the minimum amount of surface must be exposed to the flowing fluid; this surface, for a given sectional area, will be the least when the section is circular.

From the third law a greater amount of efficiency may be expected by allowing the water to move with a small velocity.

In decreasing the velocity at the same time to maintain a given supply, we must increase the surface in contact with the water, and thus the friction due to this cause of increased surface will be augmented; but since the resistance of friction is proportional to the square of the velocity and only as the surface, a greater advantage in this respect will be gained. From this it follows that it will be advantageous to make the pipes as large as possible, limiting their size by other considerations, such as space, expense, etc. With regard to the direction of the water, any change in the direction of the flow must be gradually introduced. The more sudden the change, and the greater the angle through which it is directed, the greater will be the opposing resistances.

Drainage-pipes should be attached to all water-ends which, by reason of exposed situation, are likely to freeze when not in use. Such pipes must be fitted to every portion of the pump in which there are water-pockets or cavities, and not to the two ends of the water-cylinder alone. Pumps in quarries and other exposed situations may have small holes drilled through the valve-deck plates and other portions of the pump, so there will be a drainage from the delivery-chamber through the pump to each end of the water-cylinder, and thus dispense with the outside pipes and cocks. These holes will take off a small percentage of pump efficiency, but their usefulness and automatic drainage will probably fully compensate for that. A pump so fitted should have either suitable priming-pipes, or a funnel attachment for filling the pump before starting.

CHAPTER VIII.

HYDRAULIC-PRESSURE PUMPS.

THE transmission of power by means of water under a high pressure is now generally employed in riveting, shearing, punching, bending, and flanging machines, as well as for the handling of Bessemer converters, lifts, cranes, and other machines in and around steel-works, ship-building, boiler-making, and other kindred establishments.

There is little or no uniformity in the pressures employed for doing the same work in different establishments using hydraulic power, probably for the reason that each management had independently worked out the necessary hydraulic detail, in most cases originating the machines; and in fixing upon a pressure to do the work it was generally without any reference to what was being done in other, and perhaps rival, establishments. It may be said that in steel-works and other metal-working plants the pressures employed for operating the hydraulic machinery are not often less than 500 pounds per square inch, and do not generally exceed 2000 pounds, although double this pressure is occasionally employed for special work.

Some memoranda relating to pressures collected in the ordinary course of business show the following wide divergencies:

Ordinary hydraulic lifts for warehouses . . .	75 to 150 pounds.
Foundry hydraulic lifts	100 " 300 "
Hydraulic cranes	300 " 1000 "
Flanging, punching, and riveting machines .	500 " 1500 "
Hydraulic shears	1500 " 2500 "
Special work ranging up to	5000 pounds per square inch.

Pressures of 1200 to 1500 pounds per square inch in metal-working establishments are probably oftener used than higher pressures, for the reason that many of the details of hydraulic machines can be better worked out for 1500 pounds pressure than for 3000 pounds, size being an important factor in many details, and especially in valve-gear, so that the ability to double the area of certain parts may be, and often is, of the utmost importance in the construction and successful operation of hydraulic machines. An instance once came under the notice of the writer in which a reduction of pressure from a proposed 4000 pounds to 2000 pounds per square inch determined the practicability of certain devices through no other reason than that a better mechanical construction could be had by making certain important and somewhat intricate detail double the area. The writer is not to be understood as saying that high pressures are not practicable, but rather that moderate pressures should be first considered before pressures of more than 2500 pounds are adopted.

In reference to the relative advantage of using a moderate pressure with a large ram in hydraulic machines, or a high pressure with a small diameter, Mr. Tweddell, an hydraulic engineer of large experience, observes that "it must be borne in mind that the friction of the water was not perceptibly increased with the increase of pressure, and consequently there was no loss in using the higher pressure, which allowed of a more compact machine; and practical experience has shown that there was no trouble in keeping pipe-joints tight at pressures of 1500 or 2000 pounds per square inch. At the same time he did not think there was any occasion for carrying such high pressures throughout the whole of a works, and a much lower pressure would be sufficient for working foundry and other hydraulic cranes; in such cases there was certainly a limit to the extension of a high pressure."

The above has reference to what may be termed live pressures; that is to say, continuous pressures from the pump through an accumulator to a system of piping leading to the several hydraulic machines, cranes, etc., and not to accumu-

lated pressures practically without motion, such as hydraulic presses and the like.

The service required of water-ends for hydraulic-pressure pumps is so entirely different from that described in the preceding chapter that it was thought best to separate the two, as they have little or nothing in common either in design or method of construction.

Plungers.—Hydraulic-pressure pumps are almost always fitted with double-acting plungers working through stuffing-boxes at each end of the pump-barrel; parallel rods couple the outside plungers to a central cross-head, to which are also secured both the inside plungers and the steam piston-rod or other driving mechanism. The stuffing-boxes should always be deep, so that a liberal quantity of packing can be inserted at one time. Plungers for small pumps working under high pressures should be made of machinery or cast steel, and finished by grinding, or by means of a lead lap, so as to insure their being perfectly round and straight. Care should be exercised in the selection of bar steel to see that it is entirely free from seams.

Plungers 3 inches in diameter and larger may be made of cast iron, turned and polished; in the event of blow-holes or other defects, they should be repaired by drilling into the plunger and the insertion of iron plugs well riveted or expanded by hammering, and afterwards finished to size. For cast-iron plungers it is customary to include the cross-head for the parallel rods in the same casting with the plunger.

Materials.—Water-ends for pressures up to 500 pounds per square inch may be made of ordinary cast iron, but for pressures greater than that, and not exceeding 1000 pounds pressure, air-furnace castings should be used; the tensile strength of well-mixed air-furnace cast iron will average not much below 45,000 pounds per square inch of section. For pressures more than 1000 pounds per square inch, open-hearth

steel castings are recommended; not that such castings are stronger on an average than air-furnace iron, but the quality of the metal is entirely different, and partakes more of the nature of wrought iron. Whatever the material used, very great care must be exercised in design that lumps and unusually thick portions do not occur, for every such occurrence means a defective casting in the central line of the junction of two or more parts. The writer is not partial to the use of brass water-ends except for small sizes, say for plungers not exceeding 1½ inches in diameter; the metal should not be much, if any, less than for cast iron to work under the same pressure, experience having shown that ordinary brass castings are liable to change shape under a continued high pressure. If the metal chosen be phosphor-bronze, it will add but little to the cost, and a much stiffer casting will be secured.

Double-Acting Pressure-Pump.—The sectional elevation shown in Fig. 129 represents the usual design for a small pressure-pump; it shows an objectionable detail in a necessity for putting the largest valves on the delivery side of the pump, where they are not needed, instead of the suction side, where they are needed. This fact does not prevent the pump doing good work, and is not considered as absolutely bad. The valve-seats are usually fitted and driven tightly into straight holes, from which they may be removed whenever desired by means of a hook bolt-head underneath the bushing, with a strap and nut above. The diameter of the upper valve-seat must be large enough to permit the passage of the lower valve through it. The valves and seats are usually of hard gun-metal or phosphor-bronze. The caps over the valves are also of gun-metal or hard brass. For small pumps no objection exists to tapping the water-end for insertion of the caps as shown, but for larger pumps, say those having plungers of 3 inches diameter and larger, square or round caps, held in place by bolts and nuts, are to be preferred.

A pressure-pump water-end having four separate valve-openings, as shown in Fig. 130, is one which admits of a

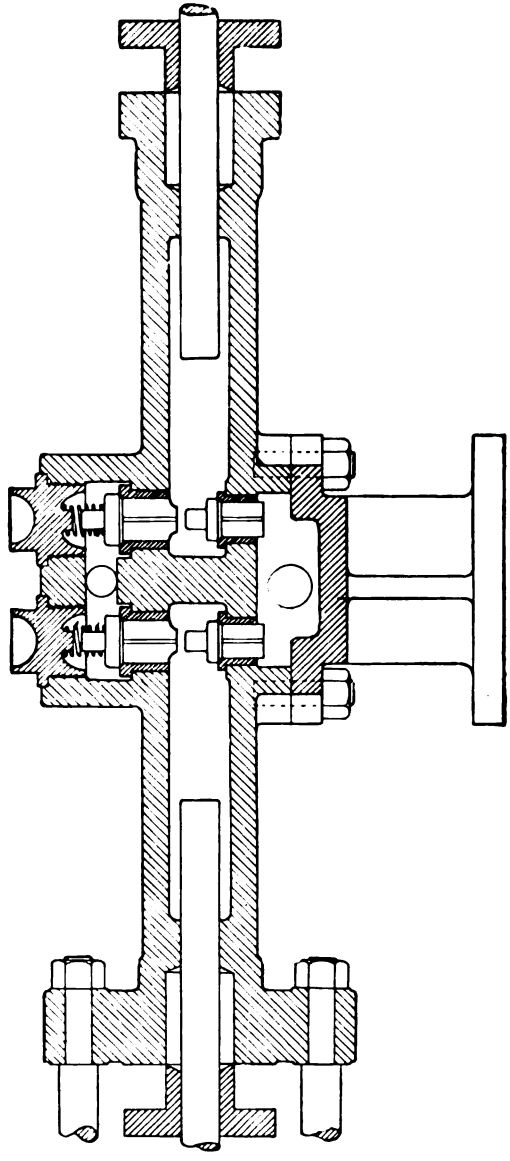


FIG. 129.

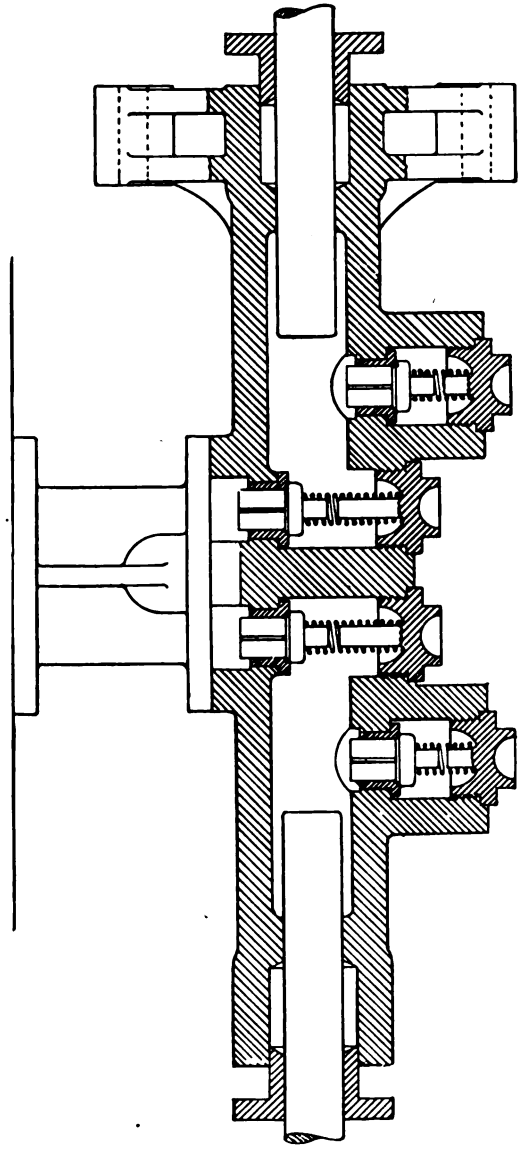


FIG. 130.

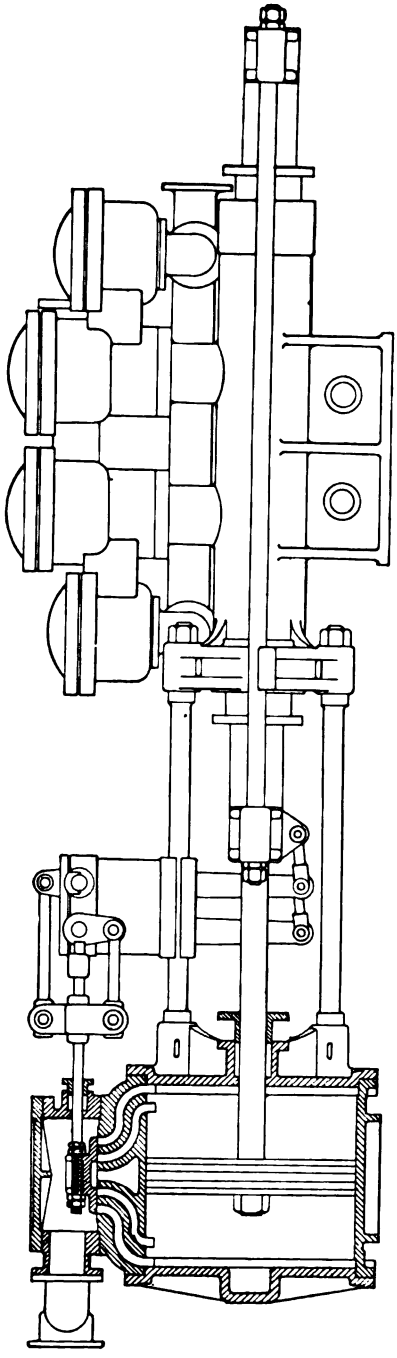


FIG. 131.

readier examination in case the pump should work irregularly, as only the valve at fault is disturbed in its seating.

A water-end with pot-valve chamber, as shown in Fig. 131, is recommended for all sizes having plungers larger than 3 inches in diameter. In this design the water-cylinder is a plain barrel, having a central diaphragm included in the main casting, with one flanged neck on each side of this partition for connecting the pot-valve chambers; the working-barrel also includes the stuffing-boxes at each end, and the connections for the tie-rods for securing to the steam-end.

A sectional elevation of a pot-valve chamber is shown in Fig. 132. It consists of two chambers, in one of which is placed the suction-valves and in the other the delivery-valves. This combined chamber

bolts to the pump-barrel at *A*. The suction-pipe bolts at *B*. The delivery of the water under pressure is at *C*. A plan of this pot-valve chamber is shown immediately below the sectional elevation. The valve-seat for the suction-valves is shown in its place; the ribs for supporting the delivery-valve plate is shown in the plan, the arrangement for the suction-valve plate being precisely like it on the opposite side.

An end sectional elevation through the delivery-valve chamber and seat is shown in Fig. 133, and an enlarged view of a section of one of the valves, together with its cage and a portion of the seat, is shown in Fig. 134. The method of securing the several cages to the valve-seat by means of a plate is clearly shown in the several preceding engravings and in Fig. 135. The above details are from a 9 × 36 duplex pressure-pump, designed by the author for handling a portion of a Bessemer steel plant, the pressure being 400 pounds per square inch, the operation being continuous throughout the year. This arrangement of seats and valves permits the

FIG. 132.

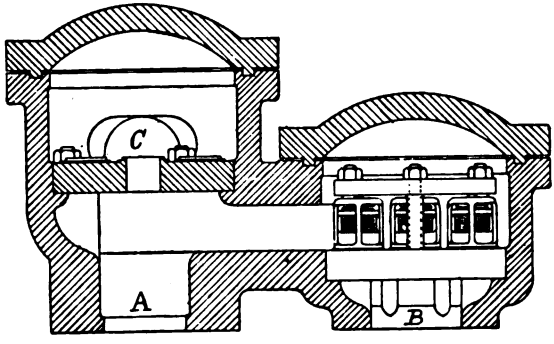
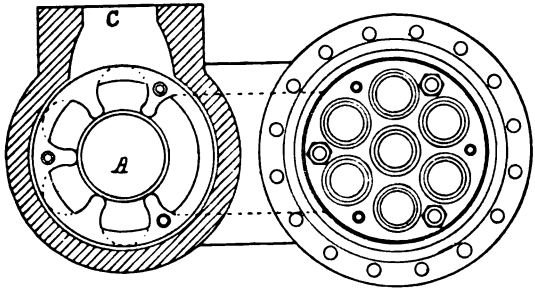


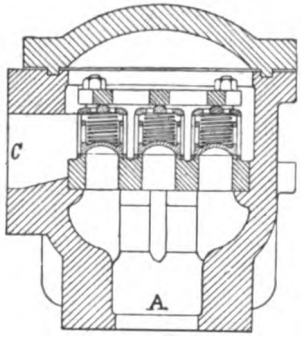
FIG. 132 A.



Plan of Fig 132.

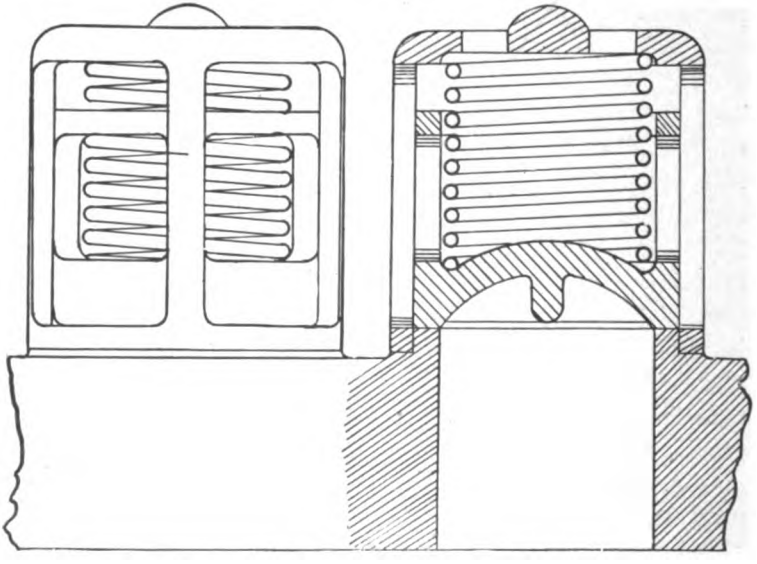
removal of one set of valves and the substitution of another without stopping the pump but the few minutes necessary to make the change. In fitting pot-valve chambers, valves, and valve-seats, everything should be made to gauges, so that similar parts will interchange throughout the series; there will then be required but one extra set of valves and one extra seat to replace any one of the eight sets included in the pump. The valves and seats are made of hard gun-metal, the springs of phosphor-bronze.

FIG. 133.



Valve-Plate and Valves for 2000 Pounds Pressure.—The valve-seat and valves shown in Fig. 136 are from a

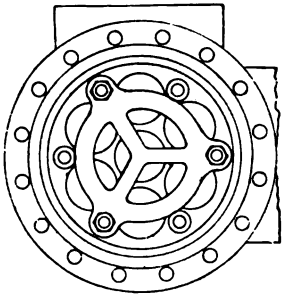
FIG. 134.



smaller pump of the same general design, working under a pressure of 2000 pounds per square inch. The valves are 2

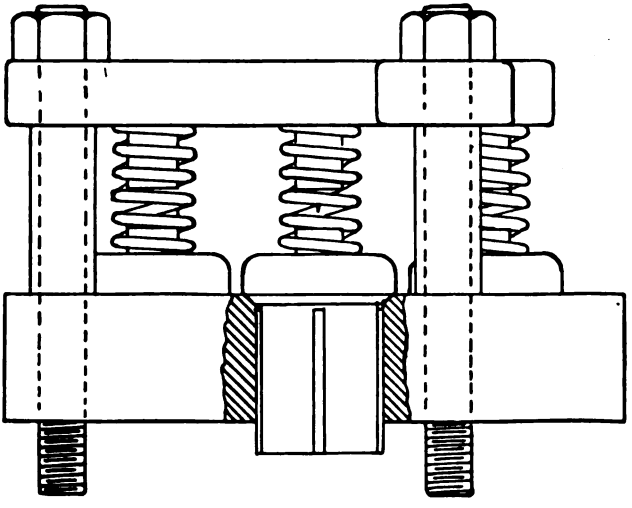
inches in diameter, mitred as shown in the drawing, and provided with four wings each; this same arrangement has been applied to a pump working occasionally, but not continuously, under 3000 pounds per square inch.

FIG. 135.



Pot-valve chambers are sometimes made to contain but one suction- and delivery-valve each; if the size of the pump is such that but one pot-valve chamber is required for each end of the working-barrel, no objection exists to applying it instead of the insertion of the valve-seats in the working-barrel, as shown in Fig. 130; but when two or more such pot-valve chambers are required, the writer recommends that the pot-valve chamber be made to include seats adapted to whatever

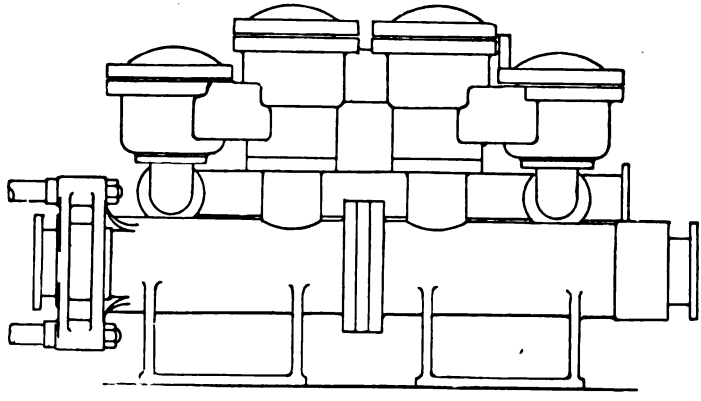
FIG. 136.



number of valves may be necessary, and thus keep down the number of separate parts requiring to be bolted to the working-barrel.

Pressure-pump water-ends of large size ought to be cast in two pieces and bolted together with a blank flange between the two ends, as shown in Fig. 137. This particular water-end is the one referred to in the description of the pot-valve chamber details given above. In large water-ends there is always a chance that the metal diaphragm between the two working ends of the barrel will be defective by reason of the

FIG. 137.



difficulty in getting a proper vent for the cores at that point; there will also be required chaplets for securing the core in place, and these of themselves interfere more or less in securing sound castings. It is a very serious matter to lose a casting of a large water-end, especially if the metal is 2½ to 3 inches in thickness, as the cost of breaking is sometimes quite as great as would purchase the same weight of pig-iron.

Accumulator.—It is the common practice in hydraulic work to have somewhere in the system an accumulator for storing up a convenient volume of water under pressure. This accumulator serves a useful purpose in the fact that it renders the action of the pump less irregular than in a direct service, and in many operations such a service would not be permissible at all; therefore an accumulator may be con-

sidered as a necessary adjunct to a pressure-pump, and its influence on the pump must be taken into account. The use of an accumulator is a direct benefit in this: it is more economical to employ a small pump working under a uniform pressure all the time than a larger pump only a portion of the time. The draft upon the water-supply in any hydraulic system is always sudden, and sometimes very great if several machines be thrown into operation at the same time. It is this sudden draft upon the water-supply that causes the accumulator to descend rapidly, and as the flow is now as suddenly stopped as it was originally started, the effect of its sudden arrest may be easily imagined.

In regard to the pressure in an hydraulic riveter, shear, or other machine having a direct connection with an accumulator, it is not uniform throughout the stroke; in some respects this is advantageous,—in riveting, for example, advantage has been taken of the impulse of the falling accumulator at the end of the stroke, so that a dead pressure of 40 tons was increased to nearly 60 tons at the time of closing the rivet; the machine thus starting with the lower pressure arrived at the higher pressure at the point at which it was required.

This fact has a very important bearing upon pump-design, and is one often overlooked. If it were a gradual increase of pressure from end to end it would not be so bad, but it comes more in the nature of a blow than that of gradual increase, because it is the sudden arrest of the falling accumulator at the bottom of its fall, the effect being precisely the same as any other falling weight moving at the same velocity. Water being practically incompressible, the shock is transmitted throughout the whole system, including the pump.

It is just such experiences that lead one to distrust designs of pumping machinery based on ordinary factors of safety, especially if the water-end of the pump be made of cast iron, the accumulator one of small diameter and considerable height, coupled with the not unusual circumstance of being scarcely large enough for the work. There is no certainty whatever as to the solidity of cast iron at the centre of the

juncture of two or more pieces, and especially if these pieces are of unequal thickness. The common defect being a drawing away of the outer skin of iron from the centre and forming a cavity within. If this cavity have no outlet it is, nevertheless, a weak spot in the pump because of the reduction in area; if, however, an opening extend from this cavity into the working-barrel, or into the delivery-chamber, then every stroke of the pump and every arrest of the fall of the accumulator is an effort to split the pump open, which in time is almost sure to occur by cumulative effect, the granular structure of cast iron favoring such rupture.

Air-Chambers for Hydraulic-Pressure Pumps.—It is not a common practice to supply pressure-pumps, or systems of hydraulic piping, with air-chambers instead of accumulators, but in special cases air-chambers have been used working under a pressure of 850 pounds per square inch, and supplying a large plant of hydraulic machinery in a situation where it was not convenient to introduce an accumulator on account of its weight or size. An air-compressor will be required for keeping the air-vessel properly charged. This is in the nature of a complication, it is true, but it is essential to the proper working of such a plant.

Hydraulic Transmission of Power.—Water under pressure may be transmitted from a central source of supply to outlying points in a large manufacturing establishment at very reasonable cost, so that for driving punching and shearing machines, or other hydraulic machine tools in out-building or in a yard, the power can be furnished at a cost much less than if a direct line of shafting were used.

As to the power saved by hydraulic transmission, Mr. Tweddell obtained, experimentally, the following results. The indicated horse-power of the engine at 50 revolutions per minute was

6.65 I. H. P. when driving the transmitting shafting alone,
3.51 I. H. P. when running light without the shafting,

leaving 3.14 I. H. P. as the lowest power required for the transmitting shafting of about 300 feet in length.

This shows that when running without doing work forty-seven per cent. of the engine-power was spent in transmission by the shafting, and a further loss has to be added for the friction of the gearing employed in getting up or reducing the speed of the several tools.

In the hydraulic transmission of power, on the contrary, the loss of useful effect between the pumps and the accumulator is very little at the usual speed of working; for with one pump working, 1694 cubic inches was the theoretical delivery of the pump for 20 strokes, and 1614 cubic inches was the actual quantity pumped into the accumulator, showing a loss of only 4¾ per cent., and with both pumps working, the corresponding quantities for 20 strokes were 3388 and 3278 cubic inches, showing a loss of only 3¼ per cent. These experiments were carefully made, the accumulator registering the exact distance it travelled for a certain number of strokes of the pump.

The loss from friction in the accumulator was found to be as follows:

1250 pounds per square inch, ascending pressure in the accumulator;
1225 " " " descending " " "

therefore, as in ascending the friction had to be overcome by the pump in addition to lifting the load, and in descending the friction has to be overcome by the load itself, the amount of friction will be half the difference of pressure in the two cases, or 12½ pounds per square inch, being only one per cent. of the power. This result applies equally to the friction in each of the hydraulic machines; and as the power is generally applied to the cutting tools direct from the ram, two per cent. represents all the loss from friction, and there remains only to add the loss due to the friction of the water in the pipes, which cannot amount to much if proper care is taken in proportioning their area and in avoiding bends.

Power for Hydraulic Engines.—The increasing variety of uses to which hydraulic power is now being satisfactorily employed shows that it is not only rapidly growing into

present favor, but promises well for the future. Recent papers and discussions on the application of electric power to drive drilling machinery at the best only leave the reader somewhat sceptical as to the economy, to say nothing of the practical feasibility of thus finishing *in situ* the great amount of drilling which cannot be done during the earlier stages of the construction of ships, bridges, boilers, etc.

No doubt, however, exists as to the practical success and the economy in working results when this drilling is done by hydraulic power. For some years past a great amount of work of this kind has been done in the French naval dock-yards by very neat hydraulic drilling-engines. The type of engine used is the Brotherhood three-cylinder, with which we are all tolerably familiar; the design, at any rate, if not with the machine itself. In one arrangement the three-cylinder engine shaft carried a bevel-pinion, gearing direct into a larger bevel-wheel driving the drill-spindle, together with the usual feed apparatus. This can be attached temporarily to a bracket for drilling work—for example, on a vise bench—away from the shops. The whole affair is so light, weighing but 60 or 70 pounds, that it can be used in connection with an ordinary hand-ratchet drill-head. Hydraulic pressure at 1500 pounds per square inch (but also sometimes at 750 pounds or 1000 pounds per square inch) is conveyed to the drilling-engine by means of special flexible copper tubing. There is no more difficulty in attaching this gear to the plating on a ship's side than is experienced in the case of hand-drilling, while the speed of working, when once fixed, is nearly 10 to 1 in favor of hydraulic work. In another application the drill-head is still lighter, the power being transmitted from the engine to the drill by means of a Stow flexible shaft. The necessary speed for the flexible shaft is obtained by gearing up at the engine-end and reducing it at the drill-end.

The working parts of the Brotherhood three-cylinder hydraulic engine consist only of the three pistons and connecting rods, one crank, and one rotating balanced valve and spindle which fits into the driver and is turned direct from the crank-

pin. There are no glands, stuffing-boxes, or oscillating joints, and the wear of all the parts is taken up automatically. The engine occupies very little space, and will work up to 200 revolutions per minute.

The duplex steam-end is the one in general use for operating hydraulic-pressure pumps. Geared power-pumps are seldom used in iron- or steel-works, but are largely used in other lines of manufacture. Pressure-pumps driven by a steam-engine in which the pump-plunger is attached directly to the engine-piston are occasionally met with; it is a good form, but its use is not general.

The especial fitness of the duplex method of operating the plungers in hydraulic-pressure pumps was probably never better stated than by the late Alexander L. Holly, in his affidavit at the time of the expiration of the Worthington patents, an abstract of which is here given:

"In all Bessemer works pumping engines for throwing large volumes of water, under 300 to 400 pounds per square inch, are required to actuate cranes, hoists, converters, and other hydraulic machinery. The hydraulic machinery is the most expensive and the hardest-worked part of the plant, and the constancy and steadiness of the pumping power is the most vital feature of the whole system of machinery. Any delay or serious fluctuation in its operation in handling fluid masses of iron and steel is fatal to the commercial success of the Bessemer process.

"The pumping engine runs from twenty to twenty-four hours per day, rarely excepting Sundays. The *strain* on the engine is constant, but the *velocity* is momentarily and suddenly varied by the demand for water from a low to the highest speed. Yet the pressure on the cranes, etc., is, and for the safe working must be, nearly constant. The heavy pressure by itself puts the machinery under a severe stress; but when this pressure is also associated with great volume, requiring very large parts, the conditions of service are more severe than any other with which I am acquainted.

"The reason why this pumping engine stands this severe service without extraordinary repairs, and the reason why it imposes no extraordinary stress on the machinery it actuates, is, that the system on which it is constructed—the movement of the steam-valve of one engine by the piston of the other engine—permits the water-pistons to stop momentarily at the ends of their stroke, thus allowing the water-valves time to seat without slamming; all this being associated with a uniform velocity of piston, and hence a uniform pressure on the water pumped instead of a varying pressure, such as is caused by an irregularly-moving piston attached to a fly-wheel. The water flows through the pump and to the cranes in a constant and noiseless stream. In all other pumps with which I am acquainted, when applied to such service, there is a violent concussion of the water, slamming of the valves, and jarring and straining of all the parts, and consequently frequent break-downs of the pump and of hydraulic pipes and machinery."

Compound Pressure-Pumps.—If the conditions of service are favorable (and this is not always the case), compound steam-cylinders, either with or without condensing apparatus, may be applied to any direct-acting pressure-pump, and thereby effect a considerable saving by getting a certain amount of work out of the exhaust steam, which would otherwise escape into the atmosphere.

Compounding is not recommended for pressure-pumps where the service is irregular; that is, where the pump works with great violence for a few minutes and then comes to a state of rest, an effect following some kinds of direct service, and always likely to occur if the accumulator is too small for the work, for the usual method of controlling the steam throttle-valve is by the rise and fall of the accumulator ram. If the action of the pump can be made continuous over long intervals of time, say several hours, then compounding is recommended, provided the initial steam-pressure in the high-pressure cylinder is not less than 75 pounds.

Pressure-pumps driven by power are often arranged with two or more plungers driven from a crank-shaft; a large plunger for filling the machine or ram quickly, at a pressure say one-fourth of that ultimately required, a pressure which will do enough work in many operations to use probably four-fifths of the whole volume of water, there remains then but one-fifth more water to be pumped at the higher pressure, if each pump acted singly, one after the other, which is sometimes, but not always, the case.

It is the usual practice in constructing such pumps to have both large and small plungers in continuous movement.

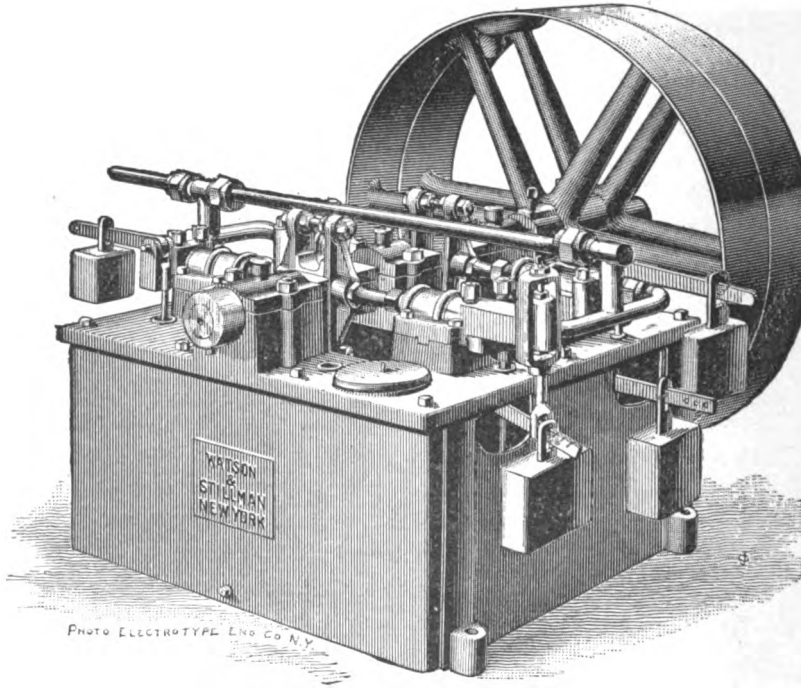
Such pumps are usually provided with an automatic device, so that when the limit of pressure for the large plunger is reached its suction-valve is lifted from the seat, and continues off the seat so long as this pressure is maintained; the large plunger during this time is simply playing back and forth in its cylinder, without doing any work; meanwhile, the smaller pump is forcing the water at the higher pressure to complete the hydraulic operation.

As a practical example, let us suppose a crank-pump fitted with two single-acting plungers of different diameters, both working at the same time, the larger one 3 inches diameter, the smaller one 1½ inches diameter, both of 6 inches stroke. Each revolution of the crank-shaft will cause the delivery of 42.4 cubic inches of water for the large plunger, and 10.6 cubic inches of water for the small plunger. If the crank-shaft make 60 revolutions per minute, 3180 cubic inches of water will have been delivered by both plungers, we will say at 500 pounds per square inch, at which pressure the large plunger-pump ceases to act by the automatic lifting of the suction-valve. The operation of the smaller plunger-pump goes on, and takes up the task of increasing the water-pressure from 500 pounds to the higher one necessary to do the work; this smaller plunger being only one-fourth the area of the larger one, will have power sufficient to force the water-pressure up to 2000 per square inch from the same crank-shaft, and at a somewhat less expenditure of power per revolution, because

the large plunger is thrown out of service; the quantity of water delivered would be only one-fifth as much; that is, 636 cubic inches per minute as against 3180 when both plungers were at work. This example assumes that two minutes are given the hydraulic operation requiring the active use of the pumps, but generally the second operation requires less time than the first, except when pressing bales, etc.

The power pressure-pump illustrated in Fig. 138 was designed by Watson & Stillman, New York, and represents one type of the medium size of hydraulic pumps built by

FIG. 138.



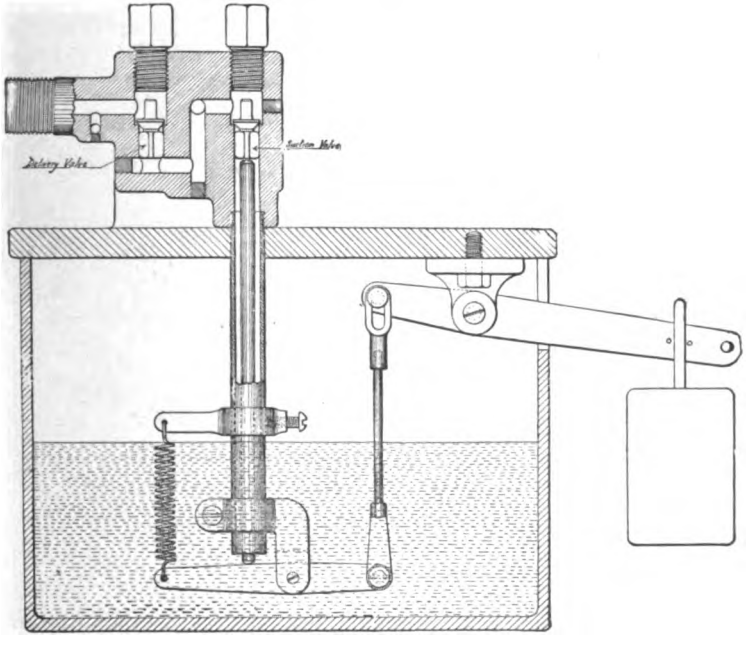
them. This particular pump is fitted with four plungers driven from the cranks and cross-heads; two of these pumps are high pressure, having plungers ¾ inch diameter, working

under a pressure of 6000 pounds, the other two plungers being 1½ inches in diameter, and working under the lower pressure of 1500 pounds per square inch; the four pump deliveries are all connected to one common discharge-pipe.

The two low-pressure pumps are provided with an automatic trip, so that when any desired pressure less than the above is reached they will automatically stop their delivery, in addition to which safety-valves are also provided for each of the above pressures; the pressure-valve of the low-pressure pumps acting as a check-valve to prevent the water of the high-pressure pumps from being driven back into the low-pressure pumps.

Although differing in design, the details of the automatic trip shown in the sectional elevation, Fig. 139, will illustrate

FIG. 139.



the principle of the method employed to stop the action of the pump. For the sake of clearness the position of the

valves were changed from that in the pump; it was also necessary to leave off the piston connecting with the high-pressure system, as shown upon the front trips in Fig. 138, but which can be clearly understood at a glance; the action being that when the required pressure has been reached the weighted lever will be lifted directly by the piston, or, in the case of an accumulator, by an accumulator trip, as in the beginning or middle of a stroke; the valve would then be seated with a heavy pressure upon it, and it would be impossible to raise it, so that a slotted cross-head must be placed upon the connection; furthermore, the spring which had been kept in a state of tension by the weight and lever is now kept in a state of tension by the pressure upon the valve only when this valve is raised from its seat by the action of the succeeding stroke. The spring prevents its reseating again by means of the lever and rod extending up through the suction-pipe, which also allows the water to be driven backwards and forwards through the suction-pipe without any loss of power.

CHAPTER IX.

STEAM AND POWER CRANK-PUMPS.

PUMPS of this type are very much in favor in Europe, and more largely made and used there than here; a circumstance probably due to the invention and development of the steam-thrown valve in this country, and now so universally employed in single steam-pumps, its use here having preceded by many years its general manufacture and sale abroad. Pumps with steam-thrown valves are compact, efficient, and offered at a lower price than is possible with fly-wheel pumps; the result is they meet with a ready sale, so that for twenty-five years past fly-wheel pumps have been offered at a great disadvantage because of its cheaper rival.

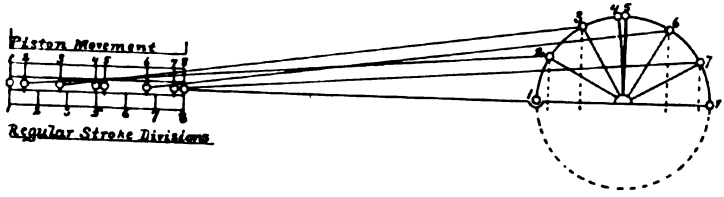
The question is sometimes now asked, though less frequently than formerly, as to the comparative merits of crank and fly-wheel as against direct-acting duplex pumps for small and medium sizes, say those having a capacity less than one million gallons in twenty-four hours. This is not an easy question to answer, because there are so many things entering into the problem which lie wholly outside of the relative merits or demerits of the two designs when brought in direct comparison.

Without entering into detail, the general conclusion reached by engineers and users of steam-pumps is that, in general, the small and medium sizes of direct-acting single and duplex pumps compare favorably as an investment, admitting the fact that they are less economical in the use of steam.

The circular motion of a crank-pin must of necessity be continuous and nearly uniform if the crank-shaft be fur-

nished with a fly-wheel of proper proportions. An analysis of the relative motions of a crank-pin and its corresponding piston, assuming the connecting-rod to be three times the length of stroke, is shown in Fig. 140. The piston movement

FIG. 140.



is not so objectionable on the first and last quarters of the crank movement, but the rapid acceleration of the piston just before the quarter stroke is reached, continuing to the half stroke, and then of its retardation until the three-quarters stroke is passed, brings some very complicated strains upon all the reciprocating parts of the pump and their connections.

The best position for the cranks on a double pumping engine is at right angles to each other; this equalizes somewhat the operation of the pumping engine as a whole, but does not remedy the faulty operation of each water-end singly.

Crank-pumps for this reason require to be stronger than direct-acting pumps of the same size; the energy stored up in the rim of the revolving fly-wheel, together with its nearly uniform rate of revolution, as well as its inability to suddenly change either of the above conditions in a single stroke of the pump, taken in connection with so inelastic a substance as water, make the conditions difficult to provide for. The difficulties which attend the practical solution of such a problem lead one easily to the conclusion that a crank and fly-wheel are not only imperfect but undesirable in a train of pump mechanism, and it is largely for this reason, as well as for cheaper construction, that direct-acting steam-pumps

have largely displaced the crank and fly-wheel pump for the commercial or small sizes.

The advantages claimed for the crank over the direct-acting movement is that the length of stroke is fixed, and no short strokes can occur, thereby securing a measured displacement for each stroke; thus crank-pumps require less clearance at each end of the steam-cylinders, and it follows also that less steam is used per stroke and with greater economy. This is true in part, and becomes so only when the engineer in charge of a direct-acting engine fails to adjust the dash-relief, or cushioning valves, so as to make a full stroke, and yet prevent the pistons striking the heads, a thing not at all difficult to do.

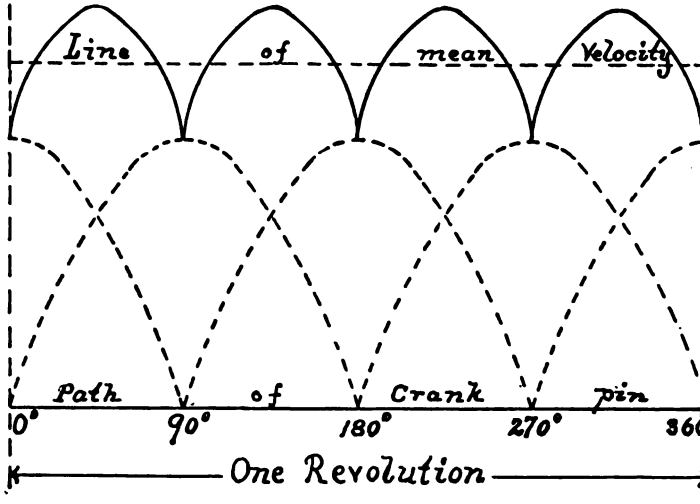
Mr. H. P. M. Birkenbine says, with reference to and in favor of crank and fly-wheel pumps, "A higher piston-speed can be had with a crank and fly-wheel pump than if the pump were direct-acting, for the reason that in the latter type the termination of each stroke is defined and secured by steam acting as a cushion to counteract the force of the moving parts and of the water and bring them to rest. In large steam-pumps 100 feet per minute may be considered as a limit to safe piston-speed. With pumping engines having cranks, connecting-rods, and fly-wheels to terminate and define the stroke of the piston any piston-speed possible to the pump can be secured with safety. The power stored in the moving mass of the fly-wheel at the termination of the stroke is carried to the beginning of the next stroke, without any loss but that due to the friction of the moving parts and the resistance of the air to the motion of the fly-wheel. Then the practically uniform speed of the rim of the fly-wheel secures the desired motion for the piston through the connecting-rod and crank of the pump by gradually retarding the motion until the point of rest is reached, and accelerating it after the piston has passed that point."

Valve Areas.—Crank and fly-wheel pumps, by reason of the irregular plunger movement, should have slightly larger

valve areas than direct-acting pumps of the same size; they should have large air-chambers, and, preferably, each water-end should have its own air-chamber. As has been already stated, the best arrangement is to place two double-acting pumps side by side, to operate as a pair, with a crank movement at right angles to each other. When properly designed, double-acting crank-pumps thus arranged have given excellent satisfaction, and have shown exceptionally high economy in the use of steam by reason of the high initial steam-pressure, and the expansion following an early cut-off, a result which has been quite impossible to obtain in direct-acting pumping engines until within the past few years, during which time ingenious and efficient high-duty attachments have been developed.

The want of approximation to continuous effort in a crank and fly-wheel pump was shown by Mr. John G. Mair,

FIG. 141.



M. Inst. C. E., London, in a paper contributed to that institution in 1886, in which he remarks upon the delivery from a compound rotative engine, with cranks at right angles, work-

ing two double-acting pumps, supposing the connecting-rod to be indefinitely long, to be similar to that shown in Fig. 141. The deliveries are added together and shown in full lines; the variation of flow in this case is sufficient to make the pressures fluctuate to such an extent that accidents are very liable to occur when working without air. Mr. Mair states that in his own practice he has met with many cases where accidents have happened to the pump-work and rising mains when through carelessness no air was in the vessel. Such a result might have been expected. The importance of a large and properly charged air-chamber attached to the delivery side of a crank-pump is now so generally understood that bad effects are sure to follow any neglect in so important a detail.

Using Steam Expansively.—In a mill engine using a high pressure of steam, cutting off early in the stroke and expanding below the average pressure necessary to do the work, the fly-wheel serves a useful purpose in absorbing the surplus energy at the beginning of the stroke, where it is not needed, and giving it out during the latter half of the stroke, where it is needed. In this manner a properly-proportioned wheel, working in conjunction with a good governor controlling the point of cut-off, will give a very even rate of rotation to the crank-shaft, varying, of course, within the limits necessary to secure a proper action on the part of the governor.

In the case of a crank and fly-wheel pumping engine as usually designed, the case is so entirely different as to almost prevent the use or application of ordinary fly-wheel formula such as would apply in the preceding paragraph, and for the reason that the plunger-rod of the water-end passes into the steam-end, connecting directly with either the high- or low-pressure piston, or if a tandem engine, with both. A pumping engine of this type is but a modified form of a direct-acting engine, because the effort of the steam-end is transmitted directly through its piston-rod, which, by continuation through a cross-head or otherwise, extends into and becomes the

plunger-rod of the water-end, controlling the movement of the plunger, making it coincident with that of the steam-piston. In this case the power of the engine does not pass through the crank-shaft and fly-wheel, as was the case in the mill engine; the connection between the steam-end and the water-end being direct, the fly-wheel absorbs and gives off only so much of the energy of the steam-end as comes through the irregular movement of the pump-plunger due to that of the crank, together with another irregular impulse upon the plunger due to the method of steam distribution.

If a pair of double-acting pumping engines were placed side by side cutting off steam at say five-eighths of the stroke, such as would be the case with the ordinary slide-valve, the machine would work without any fly-wheel at all; and such pumping engines are now in use in mines and for other purposes. Pumping engines of this type are more economical in the use of steam than direct-acting pumps, but less so than the same type of pump furnished with a fly-wheel and a better steam distribution, together with perhaps twice or three times the initial steam-pressure.

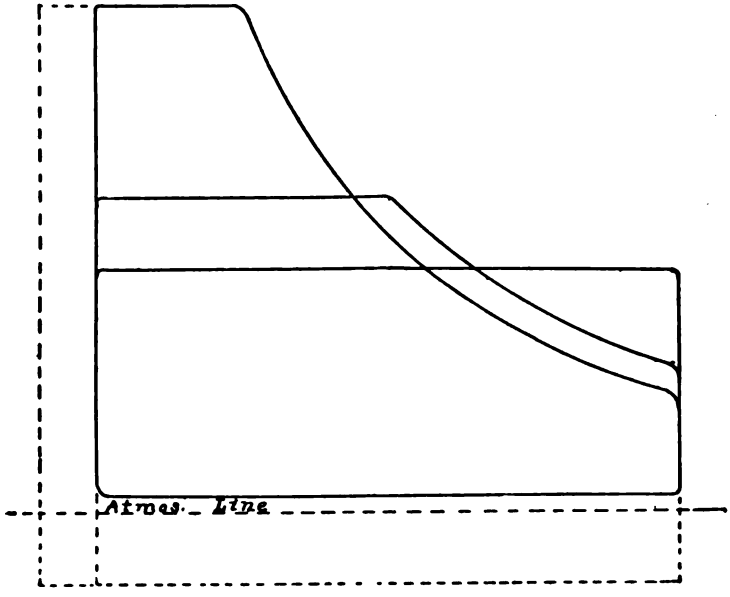
To better illustrate the best method of using steam for any purpose three diagrams are given in Fig. 142, of which two are intended to show how a saving in steam is had over a direct acting engine following full stroke, by increasing the steam-pressure and cutting off at half stroke and at quarter stroke, using the same cylinder throughout; the engine to be non-condensing, and exhausting against a back pressure of 18 pounds absolute, or about three pounds above the atmosphere. The initial pressures shown in the diagrams are 65, 80, and 120 pounds respectively.

In estimating the value of the diagrams, let us assume that the cylinder is of four cubic feet capacity, and the diagrams showing 65 pounds pressure to represent an ordinary direct-acting steam-cylinder; the initial and terminal pressures are, of course, the same, or a mean pressure of $65 - 18 = 47$ pounds. Four cubic feet of steam are used, and as one cubic foot of steam at 65 pounds pressure weighs .1569 pound, four

cubic feet would be .6276 pound of steam required to move the load one stroke.

If this same work could be done employing a higher pressure, say 80 pounds, and cutting off at half stroke, a saving

FIG. 142.



would be had as follows, using the formula given on page 181, and the table of hyperbolic logarithms on page 180.

$$80 \times \frac{1 + .6931}{2} - 18 = 49.7 \text{ pounds mean pressure.}$$

Instead of using four cubic feet of steam, only two are required. Steam at 80 pounds pressure weighs .1901 pound per cubic foot; then $2 \times .1901 = .3802$ pound of steam for each stroke, or .2474 pound less than in the preceding example, a saving of about 39 per cent.

The third diagram is that corresponding to 120 pounds absolute pressure and cutting off at quarter stroke, which represents a further saving, thus:

$$120 \times \frac{1 + 1.3863}{4} - 18 = 53\,6 \text{ pounds mean pressure.}$$

Only one cubic foot of steam is required, and this weighs .2742 pound, so that a saving of .3534 pound is had at 120 pounds pressure following quarter stroke, over 65 pounds following full stroke, a saving of 56 per cent. in weight of steam.

It will be understood that these are theoretical deductions, and that losses of several kinds which occur in practice will reduce these percentages somewhat, but they are approximately true, and will serve our present purpose, which is to show the wasteful action of direct-acting non-expansion engines as ordinarily applied to steam pumps.

Hyperbolic logarithms are seldom required in direct-acting pump calculations because the steam is not used expansively, but for convenience in such cases as involve the expansion of steam, as in the case of crank and fly-wheel, and other high-duty engines, they are exceedingly useful.

TABLE XI.

HYPERBOLIC LOGARITHMS.

No.	Log.	No.	Log.	No.	Log.	No.	Log.
1.1	.0953	3.4	1.2238	5.7	1.7405	8.0	2.0794
1.2	.1823	3.5	1.2528	5.8	1 7579	8.1	2.0919
1.3	.2624	3.6	1.2809	5.9	1.7750	8.2	2.1041
1.4	.3365	3.7	1.3083	6.0	1.7918	8.3	2.1163
1.5	.4055	3.8	1.3350	6.1	1.8083	8.4	2.1282
1.6	.4700	3.9	1.3610	6.2	1 8245	8 5	2.1401
1.7	.5306	4.0	1.3863	6.3	1.8405	8.6	2.1518
1.8	.5878	4.1	1.4110	6.4	1.8563	8.7	2 1633
1.9	.6418	4.2	1.4351	6.5	1.8718	8.8	2.1748
2.0	.6931	4.3	1.4586	6.6	1.8871	8.9	2.1861
2.1	.7419	4.4	1.4816	6.7	1.9021	9.0	2.1972
2.2	.7884	4.5	1.5041	6.8	1.9169	9.1	2.2083
2.3	.8329	4.6	1.5261	6.9	1.9315	9.2	2.2192
2.4	.8755	4.7	1.5476	7.0	1.9459	9.3	2.2300
2.5	.9163	4.8	1.5686	7 1	1.9601	9.4	2.2407
2.6	.9555	4.9	1.5892	7.2	1.9741	9.5	2.2513
2.7	.9933	5.0	1.6094	7.3	1.9879	9 6	2.2618
2.8	1.0296	5.1	1.6292	7.4	2.0015	9.7	2.2721
2.9	1.0647	5.2	1.6487	7.5	2.0149	9.8	2.2824
3.0	1.0986	5 3	1.6677	7.6	2.0281	9.9	2.2925
3.1	1.1314	5.4	1.6864	7.7	2.0412	10 0	2.3026
3.2	1.1632	5.5	1.7047	7.8	2.0541		
3.3	1.1939	5.6	1.7228	7.9	2.0668		

Table XI. contains all numbers from 1.1 to 10.0, varying by .1, which will probably meet any ordinary requirement, but should a more extended table be required, the reader is referred to Cotterill's "The Steam-Engine considered as a Heat-Engine," which contains a very elaborate table, from which the one inserted was compiled. The use of such a table in expansion of steam calculations is to facilitate and shorten the work. The hyperbolic expansion curve of steam is used on the assumption that the volumes are inversely as the pressures. This is not exactly the case, but there are so many circumstances which affect all steam-engine calculations that it is near enough true, and by reason of this close approximation and extreme simplicity, its use in connection with steam expansion calculations may be said to be universal.

The mean pressure of steam may be obtained by using the following formula:

Let p = mean pressure of steam per square inch.
P = initial pre-sure, or pressure on admission to cylinder.
R = range of expansion, or ratio of volume at end of stroke to volume at point of cut-off.

$$\text{Then } p = P \times \frac{1 + \text{hyp. log. } R}{R} - \text{back pressure.}$$

In other words, the work done before expansion is always reckoned as 1, to which must be added the hyperbolic logarithm of the range of expansion; this sum must be divided by the number of times the steam expanded, the quotient so obtained must be multiplied by the initial steam-pressure, and from the product subtract the back pressure; this will give the mean pressure throughout the stroke.

Example.—What will be the mean pressure if steam of 100 pounds follow quarter stroke or four expansions, assuming a back pressure of 18 pounds?

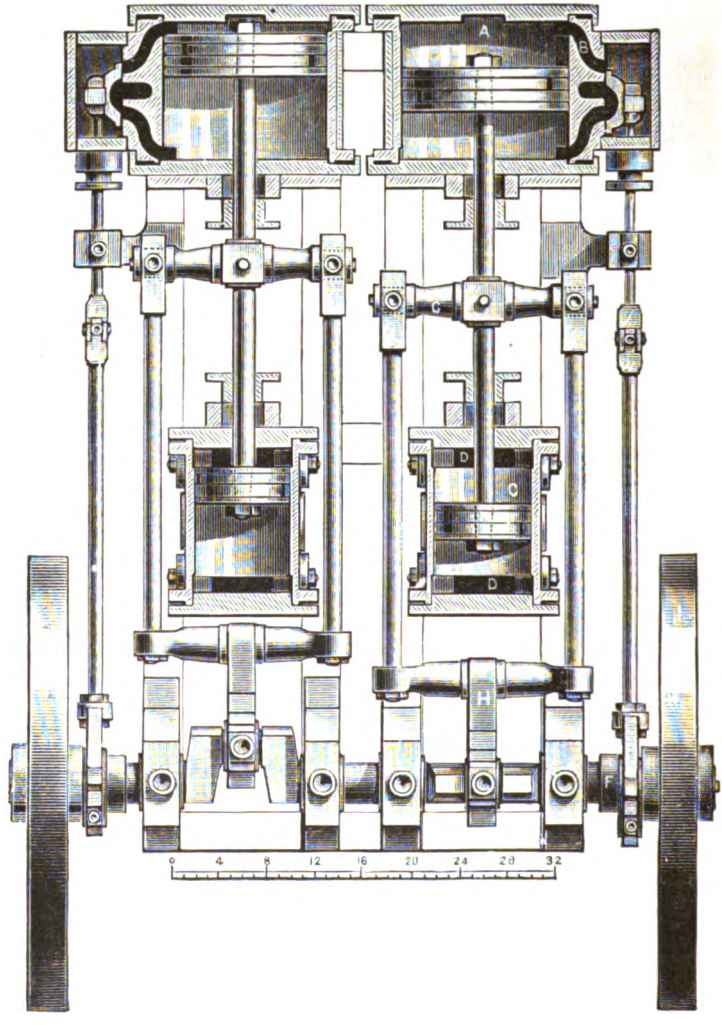
Work done before expansion = 1.
Hyperbolic logarithm of 4 = 1.386 = 2.386.

$$\text{Then } \frac{2.386}{4} = .5965.$$

$.5965 \times 100 = 59.65$
Back pressure = 18.00
Answer = 41.65 pounds.

The details of crank-pumps need not differ essentially from those of other types. It is only necessary in designing

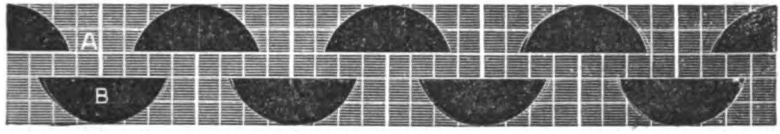
FIG. 143.



a crank-pump to select the best size and kind of water-end, and combine it with a suitable steam-engine. The completed machine may be either horizontal or vertical, depending on

the nature of the service. The examples selected for illustration show how wide a range of general design may be indulged in; some of the illustrations are much out of the

FIG. 144.

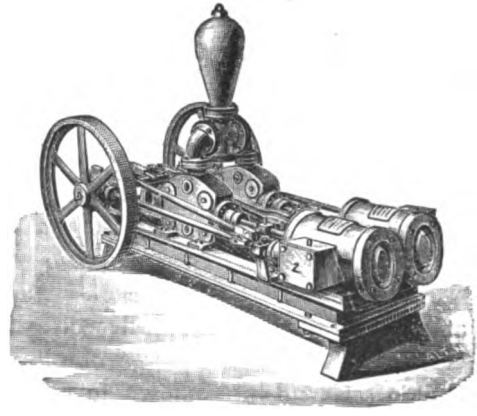


Steam per stroke, eight square units.
White lines, square of units.
White stripes, end of piston stroke.
A, cut-off or momentum; *B*, steam service.

general trend of pump design, whilst others were designed to meet certain conditions of service for which the ordinary designs were not thought to be suitable.

The Woodward steam-pump with double-acting piston water-ends is shown in Fig. 143. It consists of two complete pumps placed side by side with cranks at right angles to each other, connecting by means of suitable rods to the piston-rod of each engine. The cylinders are of the ordinary three-ported variety, the distribution of steam being effected by the ordinary D slide-valve driven by an eccentric for each engine. Two fly-wheels are used by which a regular rate of rotation is secured. The manufacturers of this pump publish a diagram, which is here reproduced in Fig. 144, showing

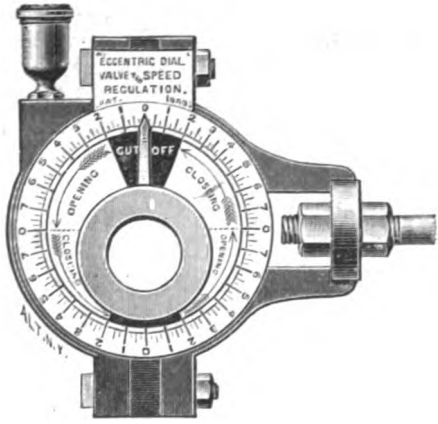
FIG. 145.



the steam consumption and distribution for the double pump illustrated above.

The cut-off is fixed at 5/8 of the stroke from the beginning, and by a proper adjustment of the slide-valves no live steam is needed for cushioning.

FIG. 146.



A perspective view of this pump is given in Fig. 145, in which it will be seen that the steam-cylinders, water-cylinders, and crank-shaft are all on the same horizontal plane.

For changing the point of cut-off, or the speed of the pump, an adjustable eccentric is employed, as shown in Fig. 146.

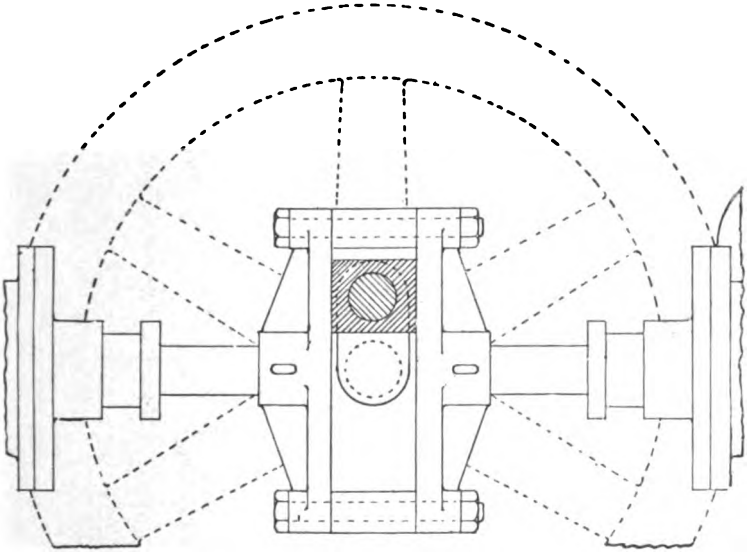
Slotted Cross-Head.—A crank and fly-wheel pump fitted with a slotted cross-head for operating the fly-wheel is shown in Fig. 147. The crank-shaft has two bearings, one on either side of the piston-rod; the slotted cross-head is fitted with a sliding block, and any adjustment for wear may be had by reducing the distance-pieces through which the bolts pass. This is the most compact arrangement yet devised for pumps, the whole length required being but little more than that necessary for the path of the crank-pin and the boss of the crank to which it is attached.

If two such pumps be placed side by side with the cranks at 90° to each other they will work without a fly-wheel.

An excellent design for a crank and fly-wheel pump by Guild & Garrison is shown in Fig. 148. The steam- and water-cylinders are mounted on a bed-plate with the bearings for the crank-shaft back of the water-cylinder. The steam- and water-pistons are connected by a rod to which is

also secured the cross-head for operating the crank. The distribution of steam is effected by an eccentric operating a plain slide-valve. The fly-wheel in a single pump requires to be heavier than when two pumps work together at right angles if any benefit is to be derived from the expansion of steam.

FIG. 147.



A double-acting vertical pump of English design (Goodbrand & Co., Manchester) is shown in Fig. 149. The method of operating the crank is a good one, because there are no strains outside of the central line of the steam- and water-pistons. The length of the connecting-rod, together with the distance required for the yoke connecting the steam and water piston-rods, gives the pump considerable height; the fly-wheel shaft is also rather high because of the yoke connection. This increased distance is less objectionable in a vertical than in a horizontal pump. No power is transmitted through the crank-pins to and from the fly-wheel other than that necessary to equalize the speed after the point of cut-off, so that the work required of the fly-wheel is very light.

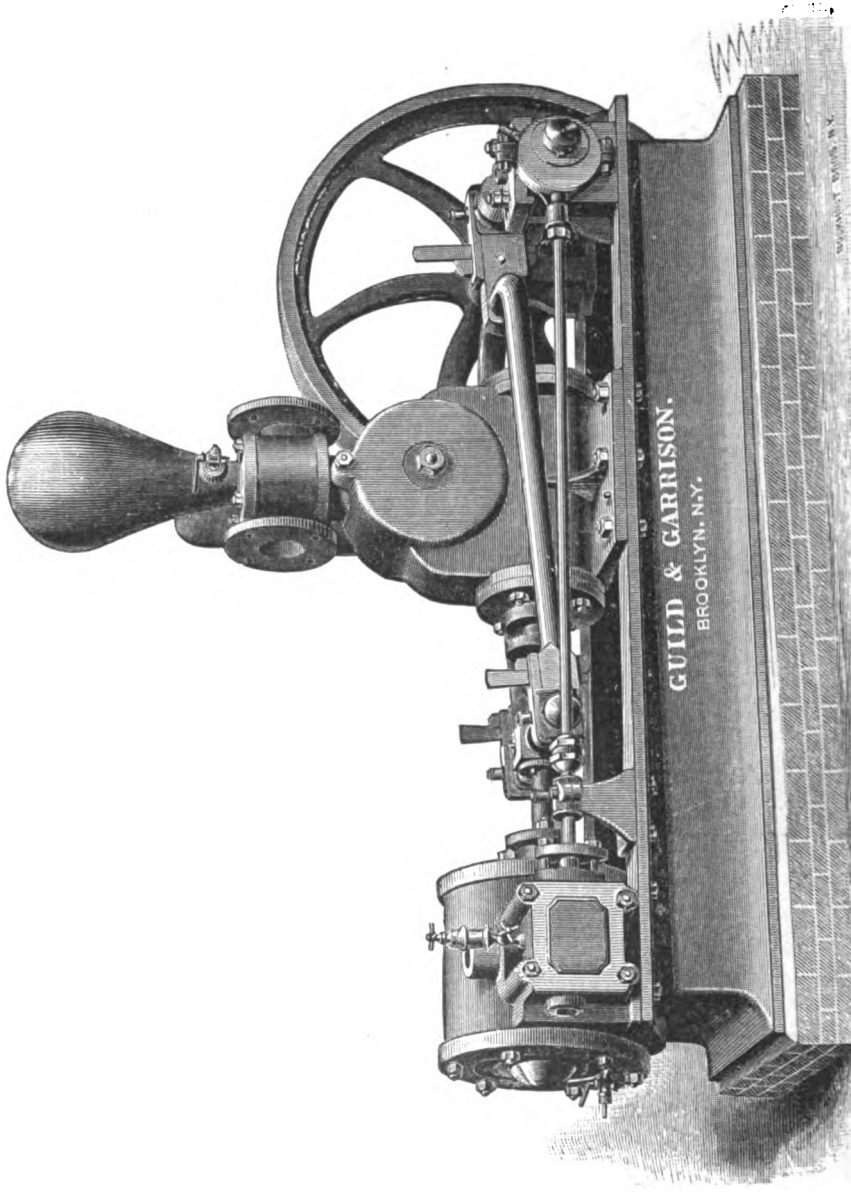
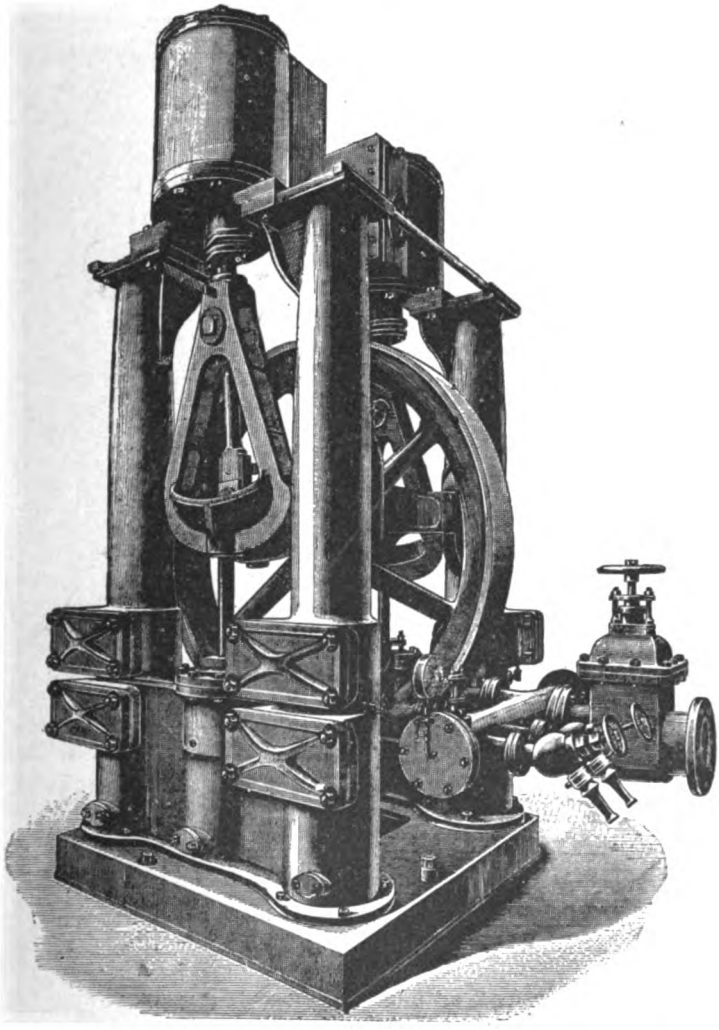


FIG. 148.

This pump has been designed to do light as well as heavy duty, and is arranged with a movable eccentric to vary the

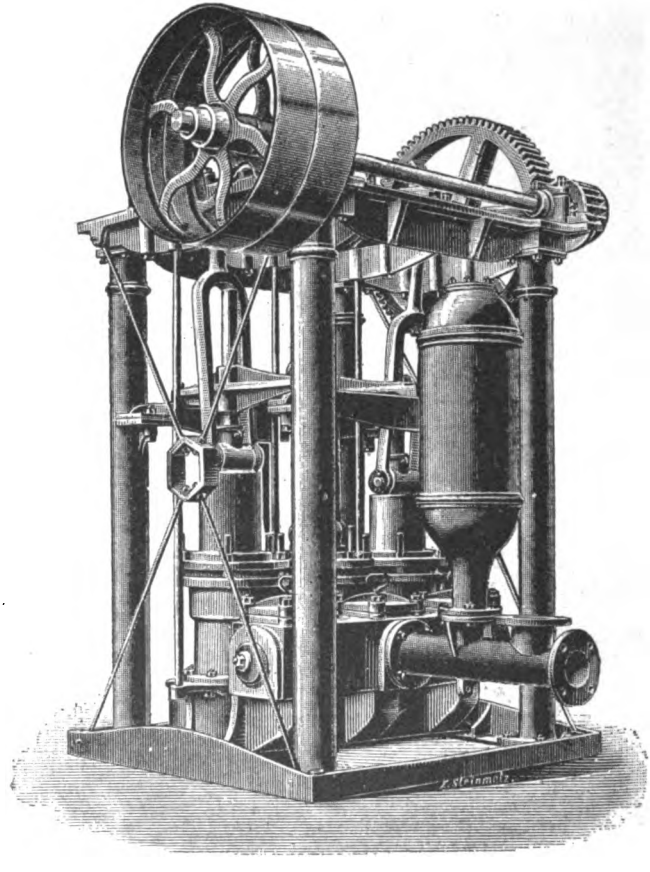
FIG. 149.



cut-off in the steam-cylinders, ranging from one-eighth of the stroke for light duty, to following the piston to about eight-

tenths of the stroke. The eccentric as described by the makers, unaccompanied by drawings, is constructed as follows: A disk, having ears as guides for the eccentric, is keyed to the

FIG. 150.



crank-shaft, and the eccentric itself is simply a ring to receive the clips for coupling up to the valve rod; it is made with two lugs to work freely in the eccentric-groove, turned on a hand-wheel, which is free to turn round on the crank-shaft, and when turned in a direction indicated by the arrow on the

disk, will alter the throw of the eccentric, suiting the duty of the pump with an economical use of steam.

Slurry-Pump.—A pumping machine for the somewhat unusual service of handling brick earth is shown in Fig. 150, which is reproduced from *Engineering*. There are three pump-barrels, each 10½ inches in diameter by 15 inches stroke. The earth is moved through a 6-inch pipe nearly a mile and three-quarters long, and is mixed with double its volume of water. A series of trials have been made to find the power required for the transportation of the earth, and the results are given in Table XII.

TABLE XII.

SHOWING THE POWER REQUIRED FOR PUMPING BRICK EARTH. PUMP BY TAYLOR & NEATE, ROCHESTER. EXPERIMENTS AT BRICK-YARDS OF SMEED, DEAN & CO., SITTINGBOURNE, ENGLAND.

OBSERVED DATA.							PROBABLE DISTRIBUTION OF POWER.				PROBABLE DISTRIBUTION OF POWER REDUCED TO A MEAN SPEED OF 130 REVOLUTIONS PER MINUTE BY DIRECT PROPORTION.				
Number of Diagrams.	Speed of Engine.	Speed of Pump.	Pressure of Pump.	Weight of 10 Gallons Slurry in Pounds.	Depth of Slurry in Mill.	I. H. P. in Cylinders.	Friction absorbed by Engine Shafting and Water Pump.	Friction absorbed by Slurry Pumps.	Power absorbed by Wash-Mills.	Power absorbed by Slurry in Pumps, calculated from Velocity and Pressure	Friction of Engines, etc.	Friction of Slurry Pump.	Power absorbed by Wash-Mills.	Power absorbed by Slurry	Total I. H. P. at 130 Rev.
1	122	Off.	Off.	. .	3′.0″										
2	114	"	"	. .	3′.0″	5.5	5.5	Off.	Off.	Off.	6.24	Off.	Off.	Off.	6.24
3	133	"	"	143	4′.4″	38	7	"	31	"	7	"	30	"	37
4	124	"	"	143	4′.4″	33	7	"	26	"	7	"	28	"	35
5	130	20.8	82 + 5	143	3′.6″	56	7	8	23	18	7	8	23	18	56
6	129	20.6	70 + 5	136	. . .	52	7	8	22	15	7	8	22	15	52
7	104	16.6	70 + 5	141	. . .	49	6	6	25	12	7	8	31	15	61
8	88	14.1	75 + 5	151	3′.0″	44	5	5	23	11	7	8	34	16	65

Note.—As the horse-power at 130 revolutions is calculated in direct proportion only, it is probably considerably under the mark where the speeds were low. If the power increased as the square of the speed, 90 I. H. P. would have been required instead of 65, and this power ought to be available for emergencies.

At these trials the entire plant consisted of two wash-mills, well-pumps, and slurry-pumps, and the slurry-pumps driven by a 20 H. P. portable engine with reduced boiler-pressure. This engine was not powerful enough for the work, and was usually assisted by a 12 H. P. portable engine driving on to its fly-wheel; during the experiments, however, the 20 H. P. engine was forced and the auxiliary disconnected.

The quantity of earth delivered through the 6-inch pipe line, 2782 yards long, was equal to 1600 cubic yards per week of 60 hours, mixed with about double this volume of water. The power averaged through a number of diagrams was about 60 indicated horse-power, of which the slurry-pumps took approximately half and the wash-mills the other half, the well-pumps, which only lifted the water about 10 feet, being quite unimportant. As only one man is required to attend to the engine, the wash-mill, and the pumps, and coal is worth $4.00 per ton delivered to the clay-pit, it is obvious that the cost for current expenses (taking the consumption at 4 pounds per I. H. P. per hour) will not exceed two cents per yard of earth transported, or about one cent per ton per mile; and when it is considered that the clay would have to be washed in any case, the cost for transport comes to less than one-half cent per ton per mile, which is probably the cheapest example of land transport on record, considering the distance. Unfortunately, the conditions of depth of slurry in the wash-mill, proportion of water in the slurry, and the speed of working varied so considerably that the results were very divergent. The above may be taken, however, as fairly approximate, as eight sets of diagrams were taken, and reduced by direct proportion (for want of a better rule) to a mean speed of the average of which has been given above.

Single-Crank Engine without Fly-Wheel.—A steam-end and valve-motion for a single-cylinder crank-pump without a fly-wheel was ingeniously worked out by Shand, Mason & Co., London. This engine was designed by them to meet the objections taken to the fire-engines of this firm which carry a

fly-wheel, and in consequence of this objectionable feature the pump cannot have the valves so arranged as to be as easy of access as they should be. Referring to the steam-end only, Figs. 151, 152, and 153 illustrate the manner in which the improved valve arrangements have been carried out by which a continuous rotary motion of the crank-shaft is obtained. The crank is assisted past the dead centres without the aid of a fly-wheel by means of a piston fitted on the slide-rod, which latter is made to perform alternately the duties of a slide-rod and a connecting-rod. As a slide-rod it is impelled by the crank-shaft, whilst as a connecting-rod the crank-shaft is impelled by it. Fig. 151 is an elevation in part section, Fig. 152 is a sectional plan, and Fig. 153 an end elevation, partly in section, of the arrangement for obtaining this result.

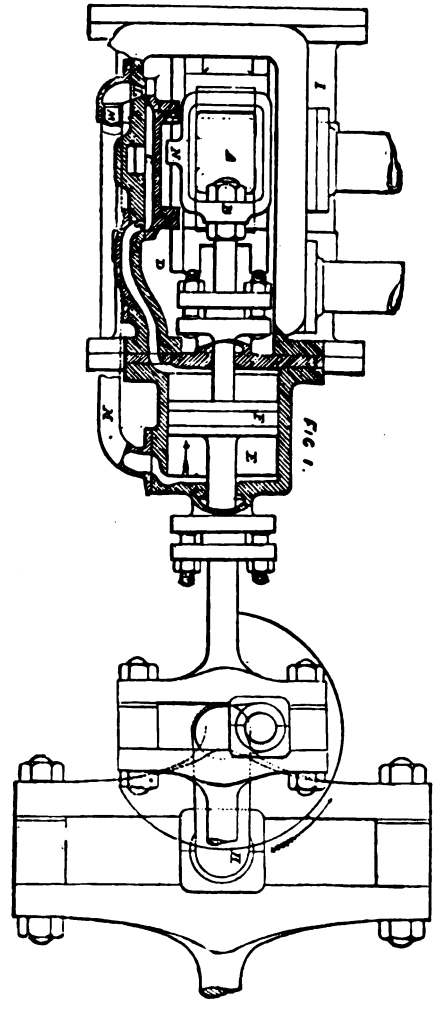


FIG. 151.

On the crank-shaft of engines in which the steam-supply is regulated by a slide-valve an eccentric is fixed for the purpose

of giving motion to that valve. The position of the eccentric on the crank-shaft is considerably in advance of the crank, and usually above 90°. Therefore, when the crank is at either of the dead centres, or at zero, the eccentric is in a favorable position for assisting the crank if auxiliary power were supplied to the slide when in this position. The slide would then become the motor of the crank-shaft, the eccentric would act as the crank, and the eccentric-rod would act as a connecting-rod. Thus, when the engine-crank is on either of its dead centres, the slide auxiliary would be in full action, and when the slide-eccentric (or crank) was on either of its dead centres, the engine-crank would be in full action. By turning these facts to account, the designers obtain a continuous rotary motion of the crank-shaft, not requiring the moving of the crank past its dead centres for starting nor the momentum of a fly-wheel

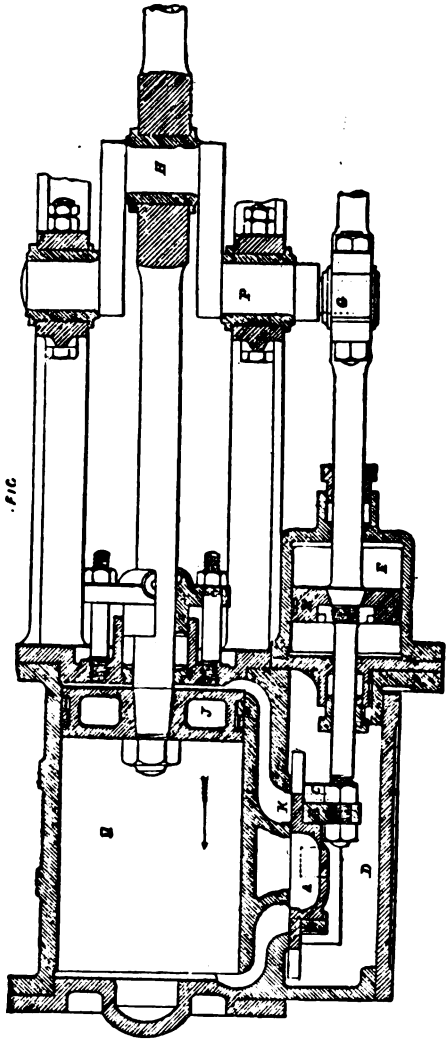
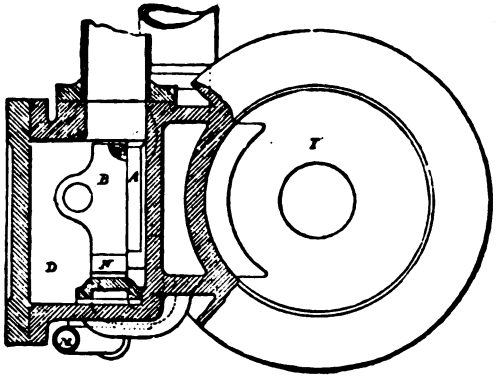


FIG 152.

while in motion, as there are no dead points on which the engine would stop, and no part of its revolution at which it would not start. To obtain this auxiliary power the slide-rod is prolonged through the steam-chest into a small cylinder. On the slide-rod and in the small cylinder is a piston, while there is provided in the steam-chest a small auxiliary slide, which is actuated by projections on it for coming into contact with a projection on the main slide-valve frame. The auxiliary valve is thus moved so as to admit the steam to, or to release it from, the small cylinder, thus actuating the piston fixed to the slide-rod, and through the slide connecting-rod or cross-head conveying motion to the small crank, and thus applying power for assisting the main crank over its dead centres.

FIG. 153.



Referring to Figs. 151 and 152, *A* is the steam-valve; *B*, the frame enclosing that valve; *C*, a small auxiliary valve; *D*, the steam-chest; *E*, a small auxiliary cylinder; *F*, its piston; *G*, a small crank for actuating the slide; *H*, the engine-crank working in a cross-head; *I*, the engine-cylinder; and *J*, its piston. The piston is shown at the beginning of its stroke, the crank *H* on the dead centre, and the small crank *G* in position for opening the steam-port *K*, and at about 100° in advance of the crank *H*. During the time the crank *H* is on the centre, steam is passing through the small port *L* and along the pipe *M* to the end of the auxiliary cylinder *E*, and pressing the piston *F* in the direction of the arrow will move the crank *H* by the crank *G* past its dead centre into a position in which it

has power. The crank H will then carry the crank G to its dead centre, and the slide A to the end of its stroke. Meanwhile, the projection N on the sling B will touch the projection O on the slide C and push it across the port L, opening the same to the exhaust, and opening the port S at the other end of the auxiliary slide, and admitting steam to the other side of the small piston F, to repeat the action on the opposite dead centres, the two cranks thus continuing to assist each other, and maintaining a continuous rotation of the crank-shaft P. The small crank G represents the usual eccentric for working the slide-valve, its action being the same; but, to avoid friction, its diameter has been reduced, and it is called a crank.

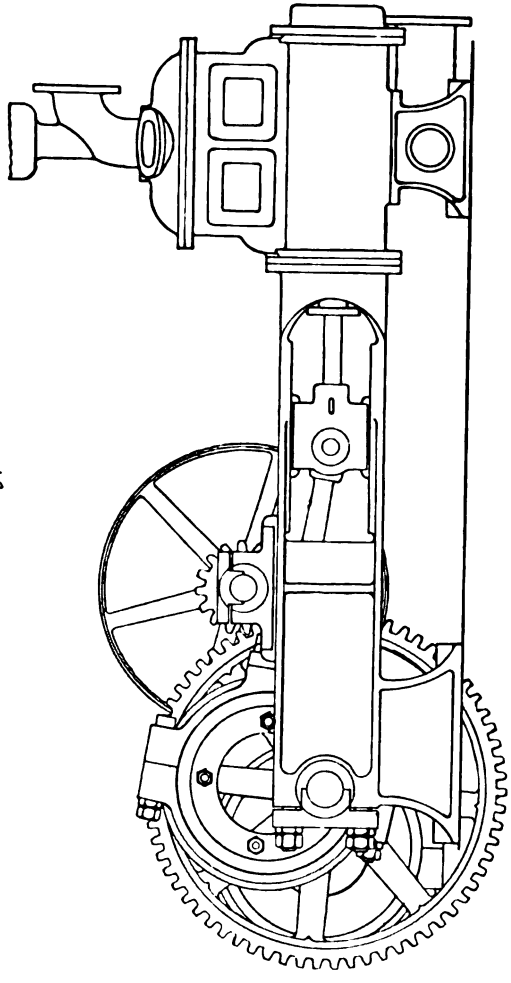


FIG. 154.

A geared pump with eccentrics adapted to a duplex water-end is shown in elevation in Fig. 154, and in plan in Fig. 155. This particular pump has a water-end 8 inches bore by 12 inches stroke, similar to the sectional elevation, Fig. 116. This pump has been working continuously for three years past at a pressure averaging more than 100 pounds per square inch.

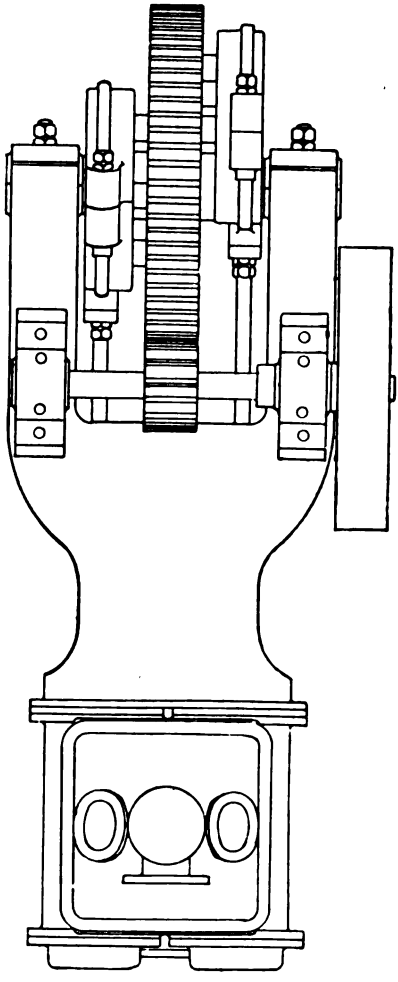


Fig. 155.

It was with some hesitancy that the writer adopted eccentrics when designing this pump, but the success attending the practical working of a number of power-pumps thus operated has completely dispelled any first doubts had regarding the employment of eccentrics for pumps of 12 inches stroke. Pump-plungers of short stroke operated by eccentrics have long been in use in this country for boiler-feeding and other purposes, and have rendered good and economical service. An advantage which such a train of mechanism offers is its compactness. In this design the eccentrics are bolted to the main gear, no power being trans-

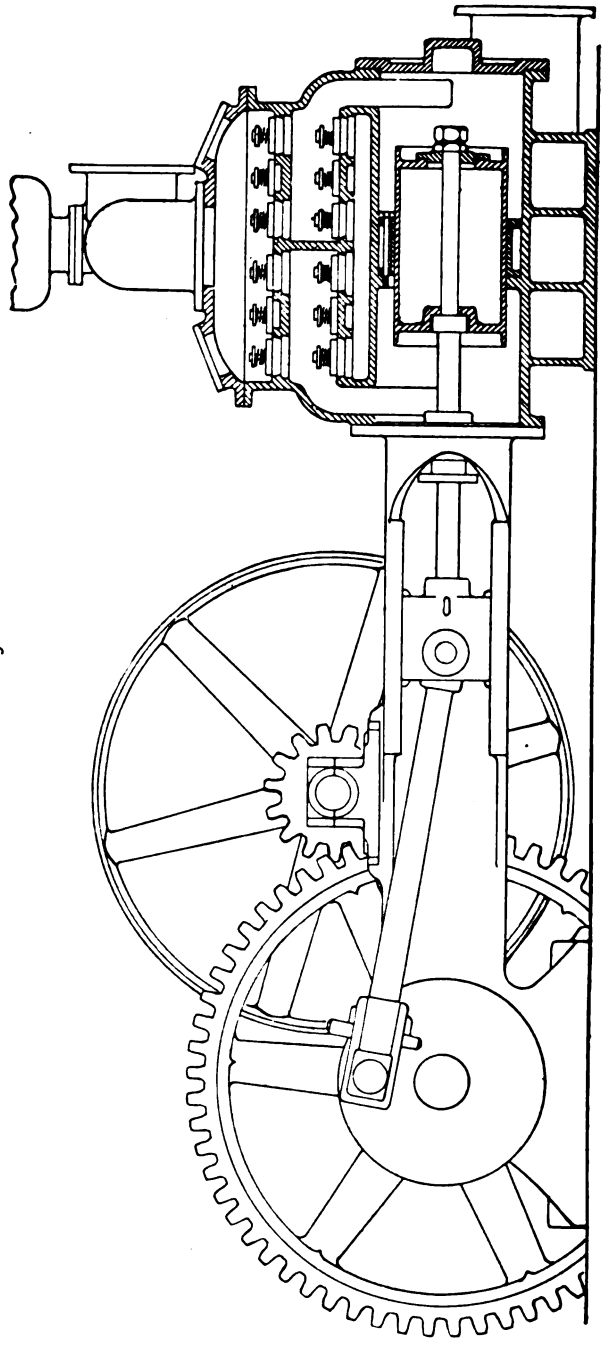
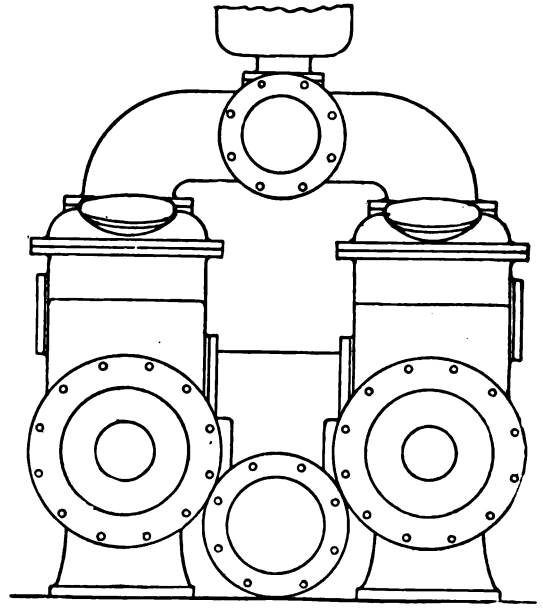


FIG. 156.

mitted through the shaft, which serves only to take the thrust of the water-plungers.

A geared crank-pump, such as shown in side elevation in Fig. 156, and in end elevation in Fig. 157, is the kind recommended for large pumps; that is to say, water-ends having plungers 14 inches in diameter by 18 inches stroke, and larger. Pumps of this kind must be made to suit the conditions of service for each installation. Greater strength is commonly given geared power-pumps than is thought to suffice for steam-pumps, and especially is this the case when a pump is to be driven by a train of gearing attached to a water-wheel. In direct-acting steam-pumps there is always an elastic cushion of steam in the cylinders with which to resist sudden jars or vibrations in the water column, but if a water-wheel be employed, its rate of revolution being so much greater than that of the pump crank-shaft, together with the fact that there is an inelastic medium at both ends of the pumping machine, unusual strains occur, for which ample provision must be made in the strength of parts; it is also the common practice to operate such pumps

FIG. 157.



at lower piston-speeds than in the case of steam-pumps. For ordinary steam-pumps the writer uses a factor of safety of ten, but for pumps of this class he does not consider it prudent to employ a factor of less than fifteen.

Electrical pumps, so called because driven by an electric motor, are now being introduced, especially in the smaller

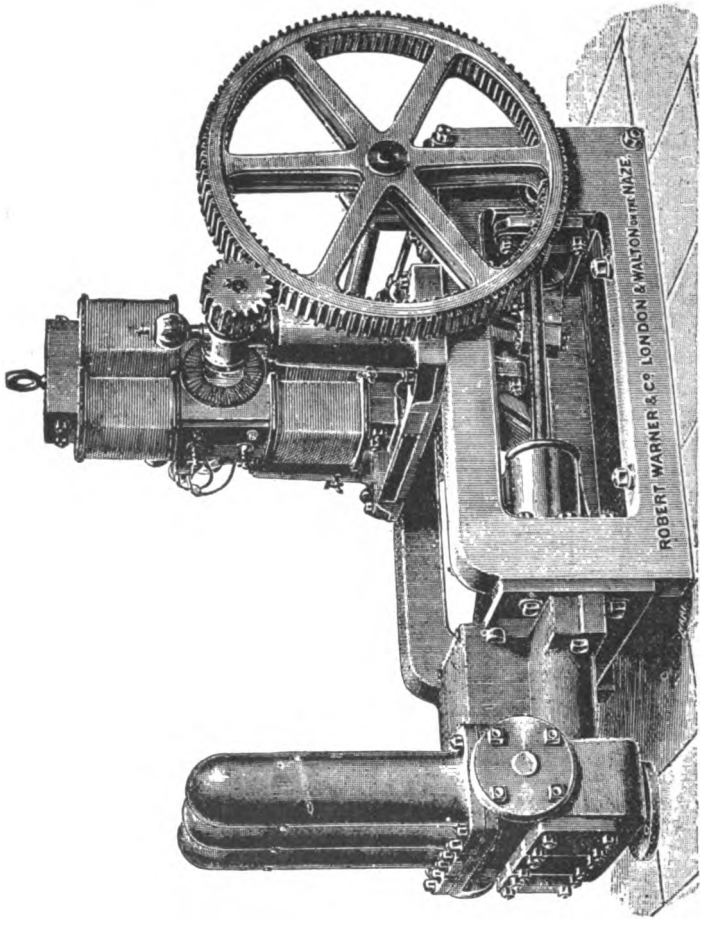


FIG. 158.

sizes, in such localities as do not readily afford either steam or belt power. The experimental introduction of electrical

pumps in mining operations promises well, but thus far no reliable data has been published in which a comparison of the best type of direct-acting engines has been brought into direct competition with electric-motor pumps when handling any considerable quantity of water, or when working under heads similar to that called for in mines. No distinct type or design of pump for this use has yet been adopted or followed by leading pump-makers, the common practice being to simply attach the electric motor to the pinion-shaft of any power-pump of a size and type best suited to the work to be done.

The illustration, Fig. 158, represents a compact and well-designed arrangement of a three-throw plunger-pump and electric motor, with double gearing to adapt the two speeds to each other. The pump is not a large one, it having a capacity somewhat less than 100 gallons per minute, the ratio of gearing being suited to a delivery of 600 feet in height.

CHAPTER X.

DIRECT-ACTING STEAM-PUMPS.

A direct-acting steam-pump is one in which the water and the steam-ends are centrally in line with each other, so that the water-plunger and the steam-piston can be attached to the same piston-rod, both working together without the intervention of a crank. This makes the simplest and most compact form of steam pumping engine, and in its application to all the ordinary operations of pumping outnumbers by far all other varieties of pumping machines combined, and this notwithstanding the fact that the ordinary direct-acting steam-pump is, perhaps, the most wasteful and extravagant form of steam-engine in use at this time. The remarkable growth of the direct-acting pump business may be explained in part by the commercial practice, which most buyers closely adhere to, and that is to pay the least possible price for a pump, just as they do for everything else. The direct-acting pump can be more cheaply built than can a crank-pump for any given capacity, and is, therefore, able to meet any possible competition in that direction; they also occupy less space than a crank and fly-wheel pump, and thus compete sharply where space is valuable enough to take that fact into account; they weigh much less than crank-pumps, so that for use on ship-board the lighter pump is generally chosen.

The practical workings of the leading designs of direct-acting pumps have, in the main, been satisfactory from the beginning; it is true that many and important changes have been made in the valve-gear from time to time, so as to give the steam-end the appearance of an entirely different design from the original, but a close examination of the fundamental

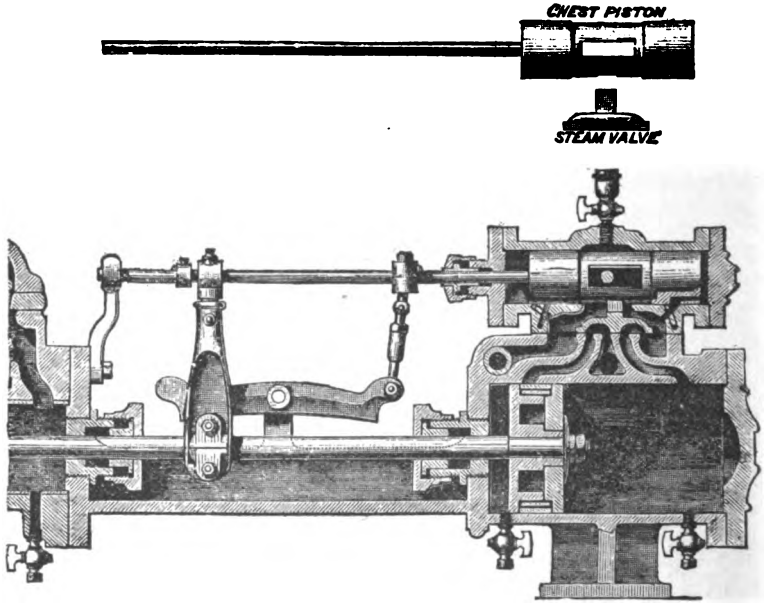
principles would show that the departure was much less than at first imagined. Two distinct types appeared about the same time nearly fifty years ago; one in which the main steam-valve was mechanically moved by a tappet-arm secured to the piston-rod, operating adjustable stops attached to the valve-rod; of this type the Worthington single pump is a familiar example to the older engineers, the manufacture of this particular design of pump having been discontinued for several years. The other type was fitted with an auxiliary steam-cylinder for operating the main valve by steam-pressure, the pistons in the auxiliary cylinder being controlled by a separate driving mechanism worked from the main piston-rod of the pump, of this type the Knowles pump is a familiar example.

The Knowles Valve Motion.—Fig. 159 is a sectional view of the steam-cylinder and valve motion of the Knowles pump. The construction is described as follows: An auxiliary piston works in the steam-chest and drives the main valve. This auxiliary, or "chest-piston," as it is called, is driven backwards and forwards by the pressure of steam, carrying with it the main valve, which valve, in turn, gives steam to the main steam-piston that operates the pump. The main valve is a plain slide-valve of the B form, working on a flat seat.

The chest-piston is slightly rotated by the valve motion; this rotative movement places the small steam-ports (which are located in the under side of the chest-piston) in proper contact with corresponding ports cut in the steam-chest. The steam entering through the port at one end and filling the space between the chest-piston and the head, drives the said piston to the end of its stroke and, as before mentioned, carries the main slide-valve with it. When the chest-piston has travelled a certain distance, a port on the opposite end is uncovered and steam there enters, stopping its farther travel by giving it the necessary cushion. In other words, when the rotative motion is given to the auxiliary or valve-driving

piston by the mechanism outside, it opens the port to steam admission on one end, and at the same time opens the port on the other end to the exhaust. Thus instant and positive motion is secured with but few working parts. There is no point in the stroke at which either the chest-piston or the main piston is not open to direct steam-pressure. A description of the operation of the pump is as follows: The piston-rod, with its tappet-arm, moves backwards and forwards from

FIG. 159.



the impulse given by the steam-piston. At the lower part of this tappet-arm is attached a stud or bolt on which there is a friction-roller. This roller, coming in contact with the "rocker-bar" at the end of each stroke, operates the latter. The motion given the "rocker-bar" is transmitted to the valve-rod by means of the connection between, causing the valve-rod to partially rotate. This action operates the chest-piston, which carries with it the main slide-valve. The said valve giving

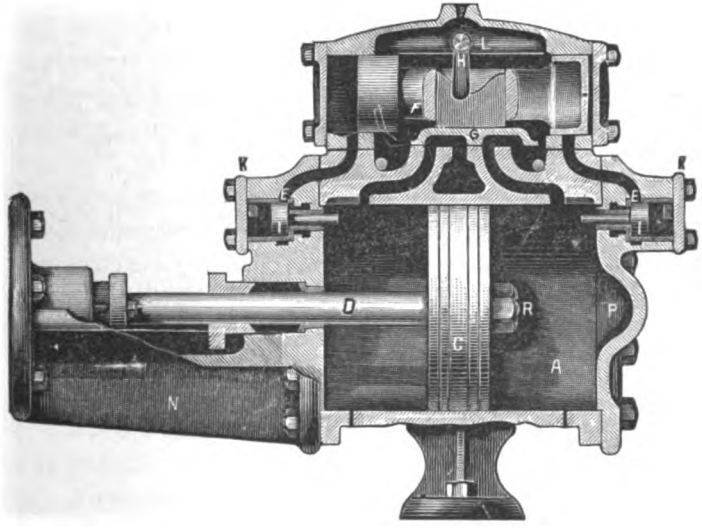
steam to the main piston, the operation of the pump is complete and continuous. The upper end of the tappet-arm does not come in contact with the tappets on the valve-rod, except the steam-pressure from any cause fail to move the chest-piston, in which case the tappet-arm moves it mechanically.

The friction-roller on the tappet-arm may be lowered or raised when desired, to adjust the pump for a longer or shorter stroke.

The Cameron Valve Motion.—A sectional view of the steam-cylinder of the Cameron pump with its valve motion is shown in Fig. 160

A is the steam-cylinder; *C*, the piston; *D*, the piston-rod; *L*, the steam-chest; *F*, the chest-piston or plunger, the right-

FIG. 160.



hand end of which is shown in section; *G*, the slide-valve; *H*, a starting-bar connected with a handle on the outside; *I I* are reversing-valves; *K K* are the bonnets over reversing-valve chambers; and *E E* are exhaust-ports leading from the ends of steam-chest direct to the main exhaust and closed by

the reversing-valves *I I*; *N* is the body piece connecting the steam- and water-cylinders.

The operation of the pump is as follows: Steam is admitted to the steam-chest, and, through small holes in the ends of the plunger *F*, fills the spaces at the ends and the ports *E E* as far as the reversing-valves *I I*. With the plunger *F* and slide-valve *G* in position to the right (as shown in the cut), steam would be admitted to the right-hand end of the steam-cylinder *A*, and the piston *C* would be moved to the left. When it reaches the reversing-valve *I*, it opens it and exhausts the space at the left-hand end of the plunger *F*, through the passage *E*; the expansion of steam at the right-hand end changes the position of the plunger *F*, and with it the slide-valve *G*, and the motion of the piston *C* is instantly reversed. The same operation repeated makes the motion continuous. In its movement the plunger *F* acts as a slide-valve to shut off the ports *E E*, and is cushioned on the confined steam between the ports and steam-chest cover. The reversing-valves *I I* are closed immediately the piston *C* leaves them, by a pressure of steam on their outer ends, conveyed direct from the steam-chest.

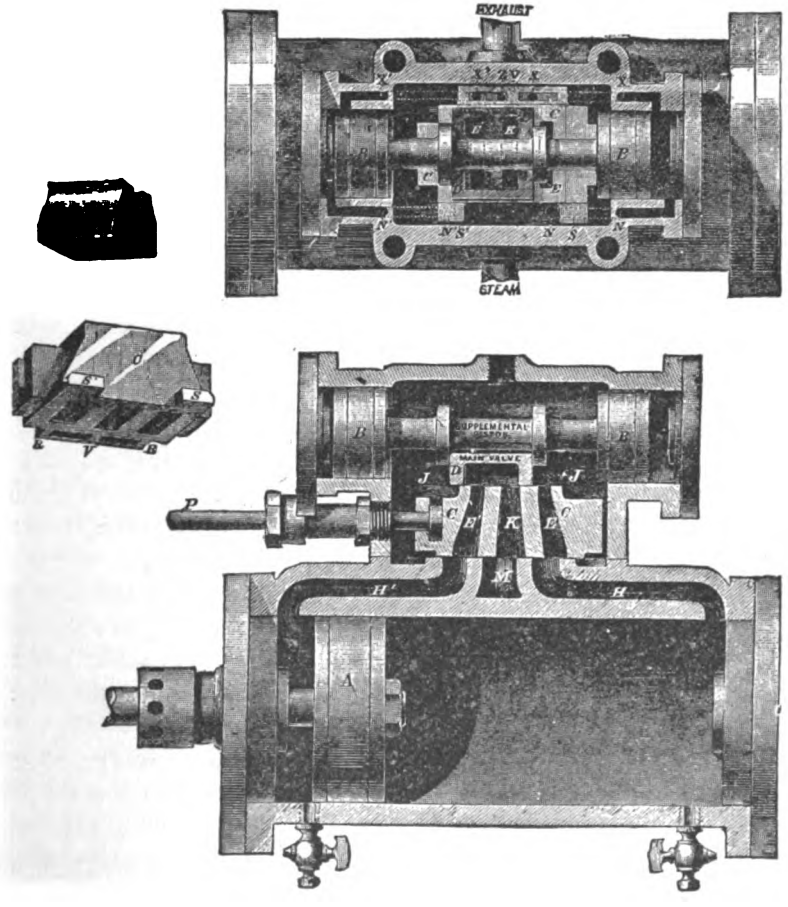
The Blake Valve Motion.—Fig. 161 shows sectional views of the steam-cylinder, valves, etc., of Blake's steam-pump. The following is the description of the operation of the valves given in the company's catalogue:

The main or pump driving-piston *A* could not be made to work slowly were the main valve to derive its movement solely from this piston; for when this valve had reached the centre of its stroke, in which position the ports leading to the main cylinder would be closed, no steam could enter the cylinder to act on said piston, consequently the latter would come to rest, since its momentum would be insufficient to keep it in motion, and the main valve would remain in its central position or "dead centre." To shift this valve from its central position and admit steam in front of the main piston (whereby the motion of the piston is reversed and its

action continued), some agent independent of the main piston must be used. In the Blake pump this independent agent is the supplemental or valve driving-piston *B*.

The main valve, which controls the admission of steam to and the escape of steam from the main cylinder, is divided

FIG. 161.



into two parts, one of which, *C*, slides upon a seat on the main cylinder, and at the same time affords a seat for the other part, *D*, which slides upon the upper face of *C*. As shown in

the engravings, D is at the left-hand end of its stroke and C at the opposite or right-hand end of its stroke. Steam from the steam-chest J is therefore entering the right-hand end of the main cylinder through the ports E and H, and the exhaust is escaping through the ports H^1, E^1, K, and M, which causes the main piston A to move from right to left. When this piston has nearly reached the left-hand end of its cylinder the valve motion (not shown) moves the valve-rod P, and thus causes C, together with its supplemental valves R and $S\,S^1$ (which form, with C, one casting), to be moved from right to left. This movement causes steam to be admitted to the left-hand end of the supplemental cylinder, whereby its piston B will be forced towards the right, carrying D with it to the opposite or right-hand end of its stroke; for the movement of S closes N (the steam-port leading to the right-hand end), and the movement of S^1 opens N^1 (the steam-port leading to the opposite or left-hand end), at the same time the movement of V opens the right-hand end of this cylinder to the exhaust, through the exhaust-ports X and Z. The parts C and D now have positions opposite to those shown in the engravings, and steam is therefore entering the main cylinder through the ports E^1 and H^1, and escaping through the ports H, E, K, and M, which will cause the main piston A to move in the opposite direction, or from left to right, and operations similar to those already described will follow, when the piston approaches the right-hand end of its cylinder. By this simple arrangement the pump is rendered positive in its action; that is, it will instantly start and continue working the moment steam is admitted to the steam-chest.

The main piston A cannot strike the heads of its cylinder; for the main valve has a lead, or, in other words, steam is always admitted in front of said piston just before it reaches either end of its cylinder, even should the supplemental piston B be tardy in its action and remain with D at that end towards which the piston A is moving, for C would be moved far enough to open the steam-port leading to the main cylinder, since the possible travel of C is greater than that of D.

The supplemental piston B cannot strike the heads of its cylinder, for in its alternate passage beyond the exhaust-ports X and X^1, it cushions on the vapor entrapped in the ends of this cylinder.

The Clarkson Valve Motion.—The longitudinal and transverse section of a direct-acting steam-cylinder, shown in Fig. 162, presents some peculiarities which are quite interesting.

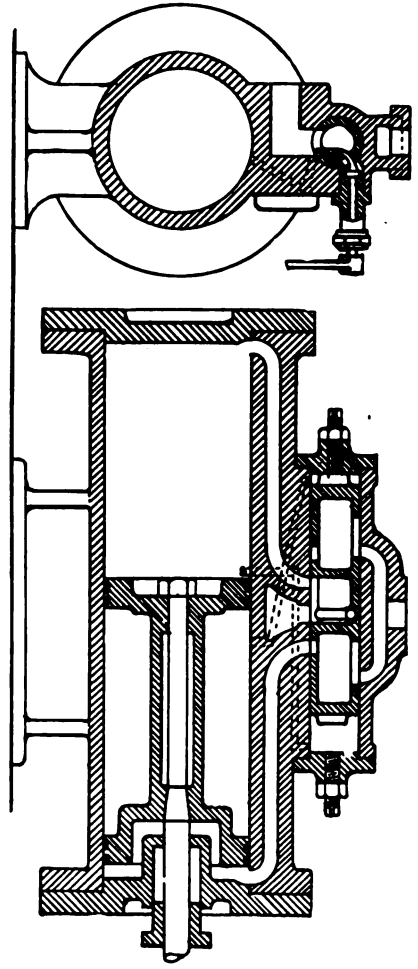


FIG. 162.

This pump is one of a large class in which the motion of the steam distributing-valve is made quite automatic, all eccentrics, tappets, or valve-gear of any kind being dispensed with. The means by which the valve is made to move are here very simple. The valve is cylindrical, and capable of an endlong motion, without rotation, steam-tight, in a cylindrical box placed on the top of the steam-cylinder. By means of closed ends and the two transverse partitions the valve is divided into three separate compartments. The steam-ports of the cylinder terminate, or rather commence, in openings into the bottom of the valve-chamber,

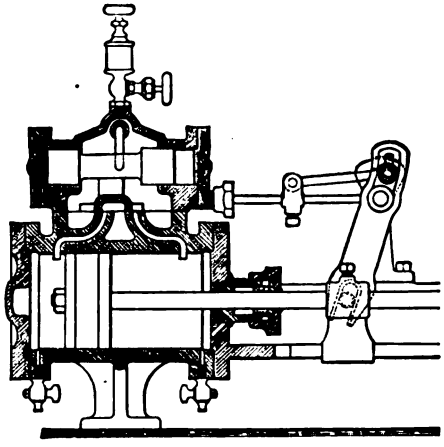
while the steam-pipe terminates in two openings in the top of the same chamber. The end compartments of the valve have holes top and bottom so arranged that in certain positions they allow free communication between the cylinder-port and the steam-pipe; the steam on being admitted to the cylinder, therefore, passes through these compartments. The middle compartment is open on the under side, and serves just the same purpose as the hollow under an ordinary slide-valve, making the communication between the steam- and exhaust-ports. The steam-piston is double-ended, the space between the two ends being always in free communication with the exhaust by means of a hole in the cylinder-wall. The clear distance between what are virtually the two pistons must therefore be equal to the stroke of the engine plus the width of the hole just mentioned. Two small holes about ⅛ inch diameter are also made in the side of the cylinder, and passages from these communicate with the two ends of the valve-chamber, each hole communicating with the end which is farthest from it. These holes are so placed that when the piston is at or near either end of its stroke one of them opens into the space between the piston (and consequently into the exhaust), and the other into the cylinder beyond one of the pistons.

We can now describe the action of the valve. When it is in the position shown, the steam passing downwards through the right-hand compartment of the valve forces the piston from right to left, the steam on the left of the piston exhausting freely through the centre of the valve. As soon as the right-hand piston passes the small hole, however, steam rushes through it, and throws the valve suddenly over from left to right (for the other end of the valve is in communication with the exhaust), and so admits fresh steam against the left-hand piston, and allows that which has just been doing the work to be discharged. At the end of the stroke the piston is again in the position shown in the engraving, the valve is thrown over from right to left, and so the operations go on *ad infinitum.*

The object of the small brass handle seen in the cross-section is to start the engine if the valves should happen to have stopped in the wrong place when the pump last ceased working.

Dean Brothers' Valve-Gear.—The motion of the auxiliary steam slide-valve is continuous like that derived from an eccentric. All other steam-pumps have an intermittent motion. The ports leading to the chest-piston are closed, except at the moment the main piston is being reversed; hence there can be no "blow through" or waste of steam in case the chest-piston becomes worn. The stroke of pump can be instantly regulated by moving the stud in slot at the upper end of lever. If raised, the pump will make shorter strokes; if lowered, it will make longer strokes.

FIG. 163.



The advantages claimed for this steam-valve gear are: It is noiseless. The auxiliary valve, having a long stroke and a rapid motion, insures a prompt reversal of the piston at the proper time. There can be no waste of steam or poor working incident to the wearing of chest-piston. It is durable and positive. The stroke of pump can be instantly changed.

Action of Valve-Gear.—The auxiliary valve F slides on the valve-seat E^2, and is provided on its under side with diagonal exhaust-cavities $d\ d^1$. The ports $b\ b^1$ and exhaust-port c are arranged in the shape of a triangle, and the diagonal cavities diverge from each other, whereby the cavity d connects

the ports b and c, and cavity d^1 connects the ports b^1 and c when the valve F is in extreme positions.

The operation is as follows: When the main piston moves from left to right, the valve F is moved in an opposite direction, opening the port b^1, admitting steam to the sub-cylinder E^1 at the moment the main piston has reached the limit of its stroke, whereby the auxiliary piston E is forced to the left, opening the main port and admitting steam to the steam-cylinder, consequently reversing the movement of the main piston. On the return stroke of the main piston the movement of the auxiliary valve is reversed, whereby the port b^1 is closed, and at the moment the main piston has reached the limit of its outer stroke, the port b is opened by the valve F, causing the auxiliary piston E to reverse its motion, opening the main port and reversing the motion of the main piston.

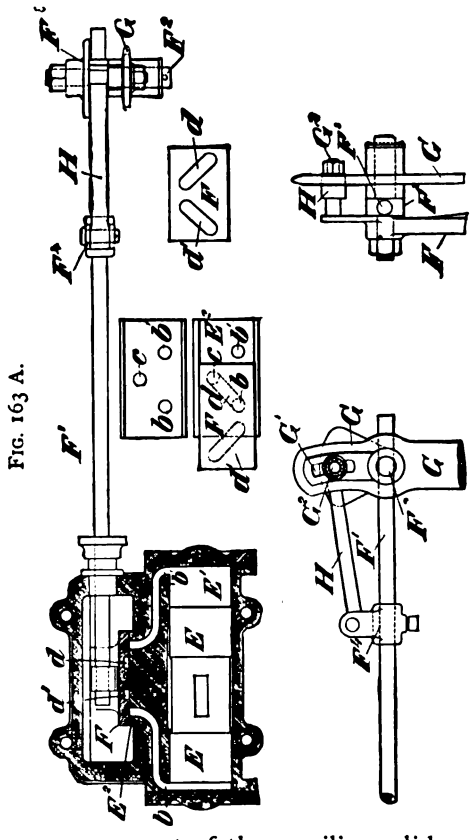


FIG. 163 A.

By this continuous movement of the auxiliary slide-valve the ports b and b^1 are kept closed, except at the end of each stroke, at the moment the main piston is reversed. This prevents any waste of steam in case the auxiliary piston is worn enough to leak as the ports b and b^1 are closed. The arrangement

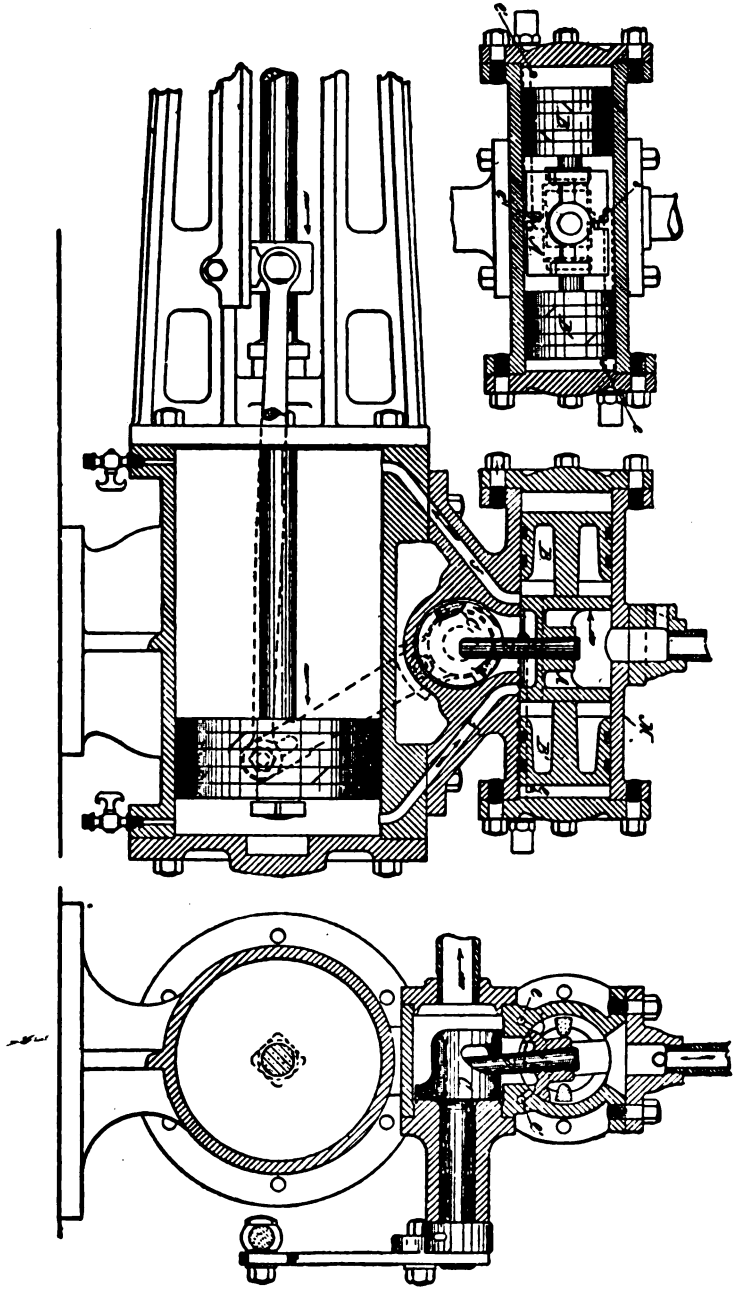


Fig. 164.

of ports admits of a short valve with a long travel. The length of stroke of pump can be regulated by moving the stud G^2 up or down in the segmental slot G^1, as this varies the travel of the auxiliary valve, thereby reversing the stroke of main piston earlier or later. The action is noiseless.

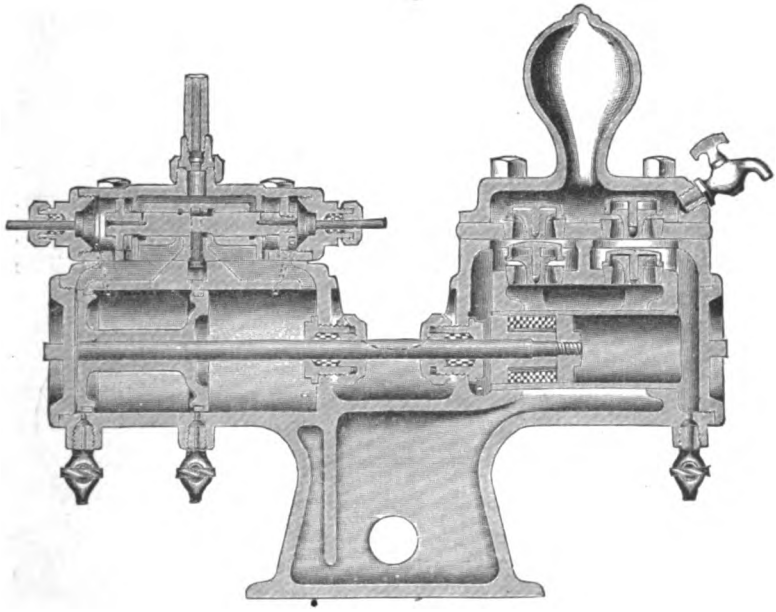
Davidson Valve Motion.—The mechanism of this valve is illustrated in the accompanying drawing, Fig. 164. Its principal peculiarity consists in the fact that it is moved both by direct steam-pressure and by mechanical connection with the main piston-rod of the engine. The cylindrical steam-chest M is bored out to make a face for the valve A and the pistons B and B', which assist in operating the valve. Sufficient space is allowed between the pistons for the valve and the steam-ports. In the exhaust-steam passage is placed a cylindrical cam, which is rocked back and forth by an arm connecting with a cross-head clamped to the piston-rod. This cam engages with a steel pin in the centre of the valve, and is of such shape that it moves the valve back and forth, and also oscillates the valve on its axis. By oscillating the valve small passages are opened and closed, opening either end of the valve-case alternately to steam and to exhaust. This gives a quick opening of the ports and largely relieves the valve motion of friction and wear.

When one of the main steam-ports f is completely open, admitting steam to the cylinder and driving main piston, cam, and valve in direction shown by arrows, the first movement of the cam will be to oscillate the valve preparatory to bringing it in proper position for the opening of the auxiliary steam-port e to live steam, and e' to exhaust. It will then bring the valve to its closure (mechanically) slightly before the end of the stroke of main piston, thereby causing slight cut-off and compression. It will then fully open the auxiliary port e to steam, and e' to exhaust. The admission of steam to one end of the valve-piston, the other being open to exhaust, throws the valve in the direction shown by the arrow, admitting and exhausting steam to and from the main cylinder for the return stroke.

The advantages of this arrangement are that the pump has no dead centre, the auxiliary ports *e* and *e'* being opened whenever the main steam-ports *f* and *f'* are closed. The closing of the valve before the piston reaches the end of the stroke eases the action of the pump and prevents the piston from striking the cylinder-heads. The fact that the valve is moved mechanically as well as by steam also insures a uniform length of stroke.

The Marsh valve-gear is shown in the sectional engraving, Fig. 165, which engraving also includes the water-valves already described. This valve movement has no external mechanism, the movements of the steam-actuated valve being controlled by the movement of the steam-piston, and con-

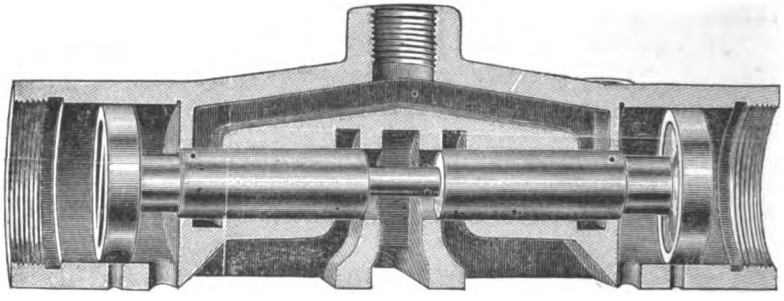
FIG. 165.



versely. A section of the steam-chest on a larger scale is shown in Fig. 165 A, and another detail of the valve in Fig. 165 B. The valve is solid and all in one piece, except that one head is screwed on, as shown in the engraving.

The steam-valve does not require setting. It has no dead centre, and will always start when the steam is admitted. The steam-piston, as shown, is double, and each head is provided with a metal packing-ring, the interior space constituting a reservoir for live-steam pressure, supplied by the live-steam pipe through a drilled hole shown by dotted lines. At each

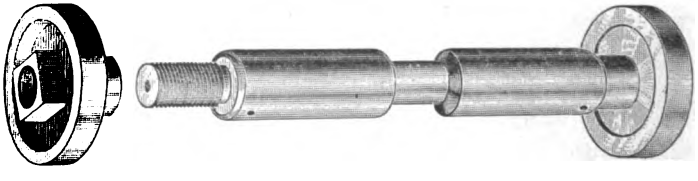
FIG. 165 A.



end of the steam-cylinder are similar holes leading to each end of steam-chest, which, together with the centrally-drilled hole and the space between the piston-heads, constitute positive means for tripping or reversing the valve with live steam.

The governing element of the valve is the enlarged heads, which present differential areas to the action of steam. The

FIG. 165 B.

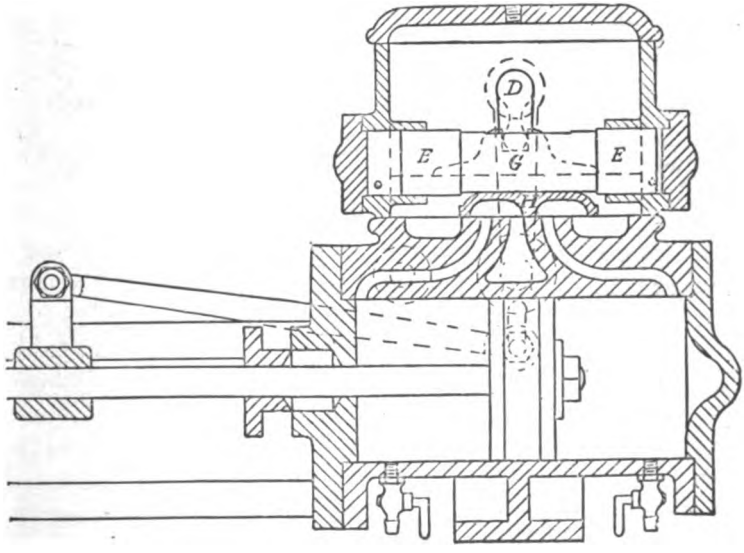


inner area of the valve-head is reduced by an amount equal to the cross-sectional area of the valve body, and is acted upon by live-steam pressure, which varies with the annular opening of the valve due to linear movement. The outer area of the valve-head is larger, and is acted upon by the pressure that moves the main piston. The working pressure in the

main cylinder acting upon the larger area of the valve tends to increase the annular opening of the valve, while the incoming steam acting upon the smaller area of the valve-head tends to reduce the annular opening. Each movement finds its limit in the preponderance of the opposing force. If the piston moves easily much pressure cannot exist in the steam-cylinder, as the piston would rapidly fly away from it, and as the pressure in the steam-cylinder and outer end of valve-chamber is always the same, the larger area of the valve-head would have slight force to enlarge the portage when the piston moved easily. On the other hand, if the piston moved hard, or against much resistance, the pressure in the cylinder would readily augment, and the larger area of the valve-head would overcome the resistance of the inner area and increase the portage of the valve.

The Guild and Garrison valve motion differs mate-

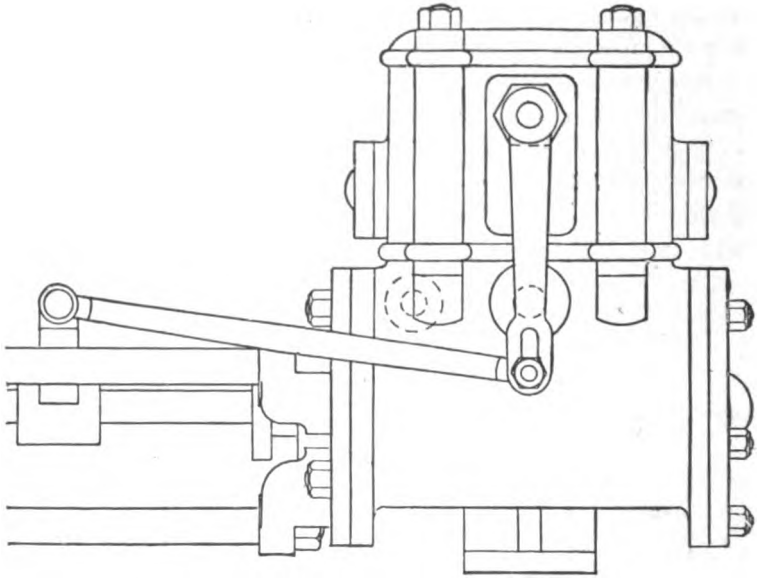
FIG 166.



rially in its arrangement from other steam-thrown valves, in that while the final stroke of the main valve is effected by a

difference in steam-pressures, the greater part of its movement is mechanically controlled by the steam-piston. A sectional elevation of this valve-motion is shown in Fig. 166, and in longitudinal elevation showing the exterior moving parts in Fig. 167.

FIG. 167.



The steam-chest is a rectangular chamber; in its interior are two small cylindrical openings, one at each end; there is also at the side a raised seat for the auxiliary valve; these are part of the casting. Small ports run from the ends of the small cylinders to the seat of the auxiliary valve and to the main exhaust-port. A plain cylindrical casting, each end constituting a piston, fits into the small cylinders. This piece, called the valve-driver, has two slots or openings at its centre, the lower one just large enough to receive a lug on the back of the main steam-valve, the top one being intended to receive the large toe of the rock-shaft. The rock-shaft crosses the steam-chest at right angles to the movement of the valves; it is made of steel and has two toes, the larger one engaging

with the valve-driver by means of the slot just mentioned, and the smaller toe fitting into the back of the auxiliary valve. There is no lost motion in the latter. Both the main steam-valve and the auxiliary valve are plain flat slide-valves, and there are no ports whatever in the valve-driver. By means of a lever and link connected with the piston-rod, any motion of the rod causes the rock-shaft to rotate and the steam-valves to move in unison.

The office of the main steam-valve, is, of course, to admit steam into, or permit it to exhaust from, the main steam-cylinder of the pump; the auxiliary valve performs the same office for the valve-driver cylinders in the chest, both being actuated by the toes upon the rock-shaft. The auxiliary valve is a D-valve, and its action is precisely the same as a D slide-valve in a steam-engine, and its effect upon the valve-driver is exactly the same as if it were a piston of a steam-engine.

In operation, the piston being at the end of its stroke and the pump about to be started, steam is admitted, and the main steam-piston moves forward; simultaneously motion is communicated to the rock-shaft by means of its connection with the piston-rod, the valve-driver and the auxiliary valve are mechanically moved at the same time and in the same direction as the steam-piston. This action continues until the piston has nearly completed its stroke, when the auxiliary valve uncovers the small steam passage leading to one of the chest-cylinders, steam enters the latter and exhausts from the opposite chest-cylinder; the valve-driver is thereby driven ahead, carrying the main steam-valve with it. The travel of the main valve is thus completed, and it is brought in position to reverse the stroke of the main steam-piston. If the pump were now stopped, the valves would be found to occupy the same relative position as at the beginning of the stroke, the valve-driver having been carried forward ready to meet the contact of the large toe on the rock-shaft upon its return.

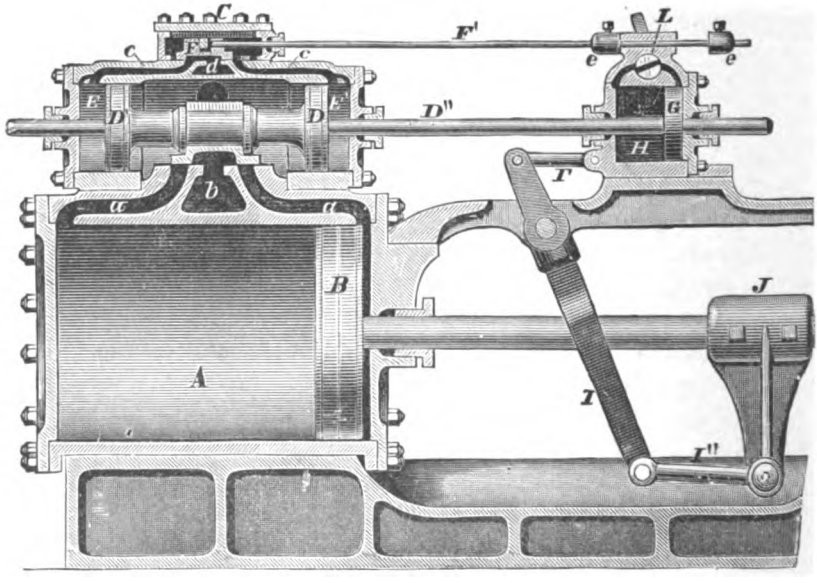
It will be seen from this description that the valve-driver, or in effect the main steam-valve, is not dependent for the greater part of its movement upon differences of steam-press-

ure, but that so long as the main pistons act it is compelled to respond to their motion; and that when steam is finally admitted to and exhausted from the chest-cylinders, this valve is already in motion, and requires but a slight additional impulse to finish its throw. It is also apparent that as the motion of the auxiliary valve is practically continuous, there can be no dead point, and that the pump will start from any part of its stroke and run with a uniform motion.

The advantages claimed for this arrangement are positive action and good wearing qualities. The construction is simple and easily understood upon inspection.

The isochronal valve-gear, by the Gordon Steam-Pump Company. Fig. 168 represents a steam pumping engine with

FIG. 168.



an ordinary steam-moved valve, the auxiliary valve being moved by a lever driven by the main piston-rod. This lever takes hold of a sliding-cylinder, *H*, whose piston, *G*, is fixed

upon a prolongation of the rod of the auxiliary piston. A cock, *L*, places the two ends of this cataract-cylinder *H* in communication, and makes the passage free or restricted as required by circumstances.

Assume the cataract-cylinder to be empty. The machine will then act as usual. At the end of a stroke the auxiliary valve will be opened, and the auxiliary piston will open the main steam-valve the full width, and so on for each stroke, the cataract-piston not interfering at all with the usual movement of the auxiliary piston. If something happens to the discharge system, such, for instance, as the bursting of a main, it is obvious that the unresisted pump-piston will be shot forward with the full force of the steam, and damage is liable to ensue. It is the office of the cataract-cylinder to prevent excessive motion of the main parts in such an emergency and to insure a uniform piston-speed.

The cataract-cylinder is filled with liquid, and it is obvious that the auxiliary piston in making its usual stroke must needs pass the cataract liquid from one end of the cataract-cylinder to the other, the cock *L* being so adjusted that this transfer of the liquid can take place just so fast and no faster.

It follows that the auxiliary piston, at proper pumping speeds, is uninterfered with by the cataract-piston, the liquid simply passing from one end of the cylinder to the other.

The cataract-cylinder is always moving, and in case the pump-piston should, by reason of resistance being removed, attempt to jump ahead, the liquid in the cataract-cylinder would have to pass much more rapidly through its restricted passages. This it refuses to do; therefore the cataract-cylinder imparts, through the medium of its liquid, more or less of its motion to the auxiliary piston and to the main valve, thus closing the main valve more or less. Briefly, if the pump seeks to move too fast it automatically affects a closure of its valve, and if the pump moves too slowly the steam-pressure upon the auxiliary piston preponderates over the cataract resistance and an opening of the main valve is effected. By this simple device complete control is had over

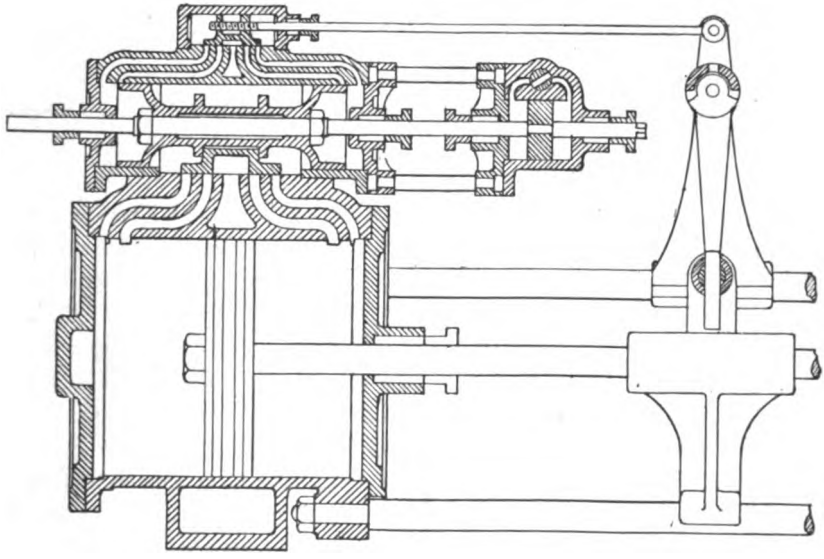
the main parts. With it we do not think it possible that serious damage could occur to a direct-acting steam pumping engine. Cases have occurred in large machines where a discharge-valve has been blown out, thus letting the full head of water against the plunger on the back stroke. This would mean destruction to an ordinary machine. The isochronal machine, as its makers aver, "continues the even tenor of its way under such circumstances." The makers test all pumping machines under steam and full load, and one of the tests of the isochronal is to restrict the discharge to the merest trifle of an outlet. While the machine is working against this immense resistance they suddenly open the discharge-valve full width, thus removing all of the resistance. The cataract-cylinder asserts itself and no running away occurs; in fact, hardly a perceptible change occurs.

Valve-Gear with Cataract.—The steam-end illustrated in Fig. 169 was designed by the writer, and exhibits the somewhat unusual feature in a steam-cylinder for a single direct-acting pump in its having five ports; but it may be explained that this valve motion was so designed that the same steam-cylinder, such as used for duplex pumps, might be used for single pumps also, the principal business at the time being the manufacture of duplex pumps. The economic advantages secured by this design in the factory consisted in the employment in common, by both styles of pumps, of the steam-cylinder, slide-valve (with slight alterations for driving it), piston, piston-rod, cylinder-heads, and stuffing-box. Instead of the ordinary steam-chest, such as usually furnished duplex pumps, a special one was made not unlike those in common use for single pumps, and which in our description we will call the auxiliary cylinder. This cylinder has a double piston, one fitted to each end, and midway between these two pistons are collars for driving the main slide-valve. This auxiliary cylinder, like the main steam-cylinder, is provided with five ports, and covering these is a plain flat slide-valve receiving its motion from a rocker-arm, the shaft of which is operated by

the swinging movement of a lever driven by the cross-head attached to the main piston-rod.

The two outer ports are for steam, the three inner ones are for the exhaust. The slide-valve has no lap or lead on either the steam or exhaust sides, consequently a steam- and its corresponding exhaust-port must be open from end to end of the stroke. When the piston travels past an exhaust-port it thereby cuts off all escape of exhaust steam from the cylin-

FIG. 169.



der, and compresses the remaining portion, which is a sufficient cushion to prevent the piston striking the heads.

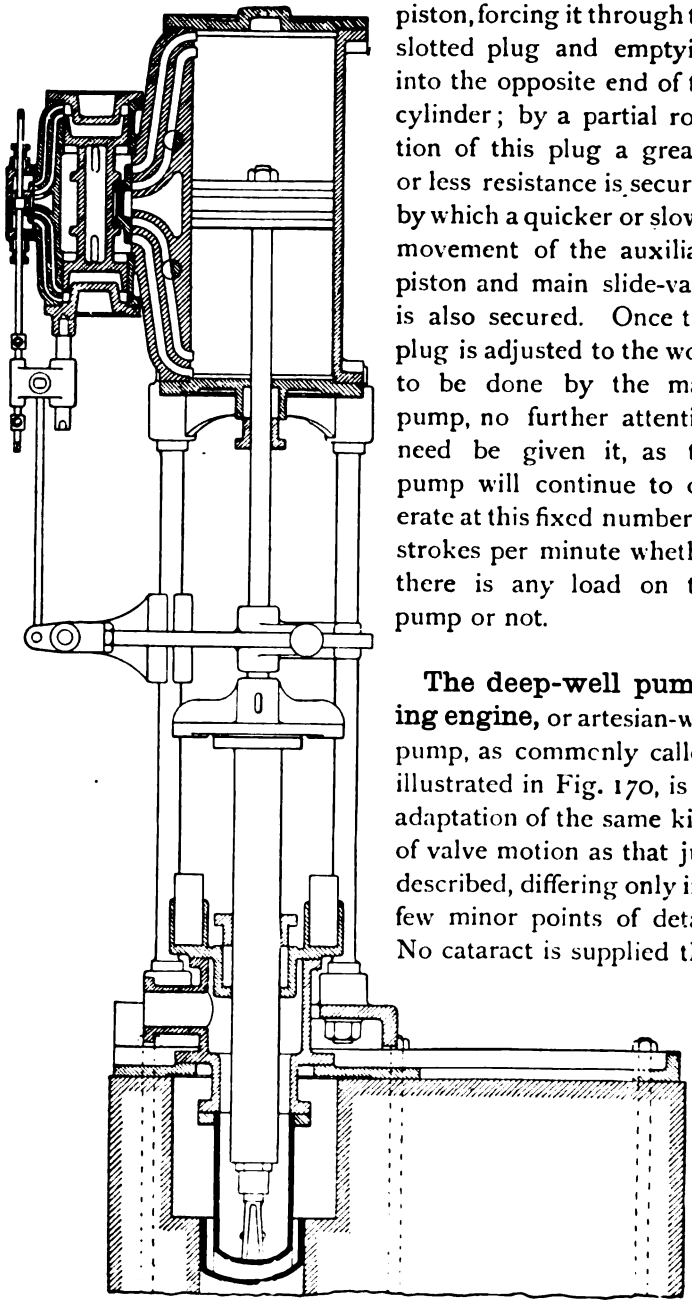
The small slide-valve above the auxiliary piston distributes the steam so as to give motion to the piston underneath by admitting steam at one end of the auxiliary cylinder, and uncovering the exhaust-port leading from the other end, the effect of which is to cause the auxiliary piston to move with the pressure, a movement which continues until its forward edge closes the exhaust-port, cushions the pent-up exhaust steam, and with the increasing pressure thus obtained the

piston gradually comes to a state of rest; at the same time that this movement is going on the main slide-valve has been carried by the auxiliary piston in the same direction, effecting a corresponding opening of the steam- and exhaust-ports in the main cylinder, and producing a similar movement of the main piston. When the main piston approaches and finally reaches the end of its stroke the vibrating lever driven by the main cross-head gives the segmental tappet a partial rotation, which through the intervention of a rocker-arm and valve-rod carries the small slide-valve over to the opposite end of the valve-chest, and the whole operation is then reversed.

The steam from the boiler is admitted to the main steam-chest; that is to say, to that space between the two auxiliary pistons in which the main slide-valve is located; as the pistons are both of the same area, they are not influenced in any way by this pressure. There is a small hole (not shown in the drawing) drilled from the steam-chest for the small slide-valve over the auxiliary pistons, down into the steam space below, so that boiler-pressure is also had for operating the auxiliary pistons. The exhaust from the small slide-valve may lead down into the main exhaust cavity, or into the atmosphere.

The adjustable cataract shown in the engraving was for the purpose of steadying the movement of the auxiliary piston. It consists of a piston and rod connecting directly to the auxiliary piston, so that their movements may be identical. The rods pass through both ends of both cylinders, so that no unbalanced pressures occur in the auxiliary steam-cylinder, and no difference in areas exists in the two ends of the cataract-cylinder. This latter cylinder has but one port extending from end to end, as shown in the drawing; between the two ends a slotted plug is inserted, by the partial rotation of which any amount of opening from a full port to absolute closure may be secured. This cylinder is to be filled with any convenient fluid, mineral oil, for example, and the plug inserted; it is then ready for service. The function of the cataract is to control the movement of the auxiliary piston and prevent a violent movement by requiring the displacement of the oil

FIG. 170.



on one side of the cataract-piston, forcing it through the slotted plug and emptying into the opposite end of the cylinder; by a partial rotation of this plug a greater or less resistance is secured, by which a quicker or slower movement of the auxiliary piston and main slide-valve is also secured. Once this plug is adjusted to the work to be done by the main pump, no further attention need be given it, as the pump will continue to operate at this fixed number of strokes per minute whether there is any load on the pump or not.

The deep-well pumping engine, or artesian-well pump, as commonly called, illustrated in Fig. 170, is an adaptation of the same kind of valve motion as that just described, differing only in a few minor points of detail. No cataract is supplied this

steam-end, as another method of governing is employed. The ports, main valve, auxiliary piston, and small slide-valve are substantially the same in design. The method of moving the tappets is different, and is probably the best device of the two, as it is easier to make any needed adjustments than with the segmental tappets.

Two dangerous accidents are likely to occur in deep-well pumping,—one, the breakage of a pump-rod, the other, a failure of water-supply. Either of these is liable to damage the steam-end, if not to wholly wreck it, especially when pumping from great depths, unless some provision be made in advance to meet such a contingency should it occur.

The cushioning of the exhaust is at once a convenient and satisfactory method of controlling the movement of the main piston. By reference to the engraving two semi-cylindrical plugs are shown in the exhaust-ports, the function of which is to control the flow of the exhaust steam from the cylinder by giving these plugs a partial rotation so as to choke the exhaust; a sufficient back pressure can be had to secure any desired rate of piston-speed, fast or slow, with a full load or without any load. As each plug may be separately adjusted, a separate movement may be secured to the upward or to the downward movement of the piston, adapting it to the depth of the well or to any other conditions of supply.

Compounding Single Direct-Acting Pumps.—When so desired, single direct-acting steam-ends may be arranged for using steam expansively by the addition of a low-pressure cylinder, tandem to that of the high. The slide-valve faces for the high- and low-pressure cylinders should be in the same plane by carrying up that of the smaller cylinder to the level of the larger. The slide-valve in the high-pressure cylinder need not differ from that which would be used if no compounding were attempted. The auxiliary piston would differ only in having a rod passing through its cylinder-head into the steam-chest of the low-pressure cylinder. To balance the pressure, an extension of the same diameter should pass

through the opposite cylinder-head; and if a similar extension be provided at the opposite end of the low-pressure steam-chest, the balance would then be complete. The auxiliary cylinder must be large enough to easily and promptly handle both slide-valves; if this cylinder be one-half the diameter of the high-pressure cylinder, ample power ought to be secured for this purpose. The low-pressure valve may be a plain slide-valve, with a suitable adjusting device for fixing its relation to the high-pressure valve, as these two valves must work together.

CHAPTER XI.

THE DUPLEX PUMP.

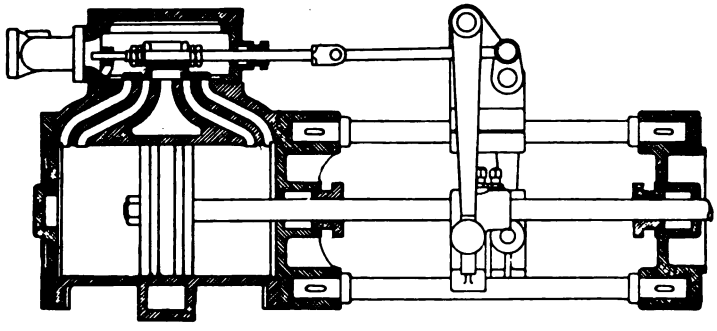
THE credit for the invention of what is commonly known as the duplex pump, distinguished for its almost ideal simplicity, with its peculiarly efficient valve motion, to which it owes its complete exemption from noise or concussive action, is due the late Henry R. Worthington.

It consists of two steam-pumps, of equal dimensions, placed side by side, with the valve motion so designed that the movement of the steam-piston of each pump shall have the controlling movement of the slide-valve of its opposite pump, the effect of which is to allow one piston to proceed to the end of the stroke, and gradually come to a state of rest; during the latter part of this movement the opposite piston then moves forward in its stroke, and also gradually comes to a state of rest; but in its movement forward, and before reaching the end of its stroke, the slide-valve controlling the first piston is reversed, and in consequence the first piston returns to its original position, and in nearing the end of its stroke it, in a similar manner, reverses the slide-valve controlling the second piston; these movements are both uniform and continuous so long as steam is supplied to the pistons.

A sectional elevation through one of the steam-cylinders of a duplex pump is shown in Fig. 171. To those not familiar with the construction of duplex pumps, a noticeable feature will be that the cylinder has five ports instead of three as in ordinary engines. The two end ports are for the admission of steam, the two inner ones are for its exhaust. The slide-valve has neither lap nor lead on either the steam or exhaust sides. The drawing shows the two ends of the steam-valve to be exactly on line with the outer edges of the steam-ports;

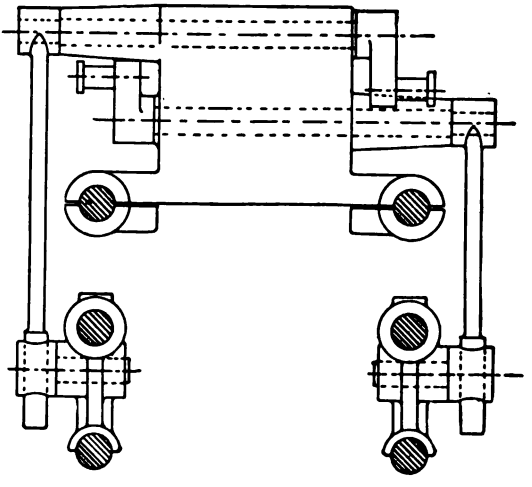
the face of the valve extends over the steam-port, the bridge, and to the inner edge of the exhaust-port, so that in its

FIG. 171.



present position no steam can enter or leave the cylinder. It will be understood that the piston movement of one engine controls the slide-valve not of its own cylinder, but the slide-valve belonging to the opposite engine; this is accomplished by means of a cross-head securely fastened to the piston-rod operating a lever, with a shaft extending across the framing of the pump, as shown in Fig. 172. The rocker-arms for moving the slide-valves are arranged one above the shaft and the other below it; this is for the purpose of securing for one engine a slide-valve

FIG. 172.

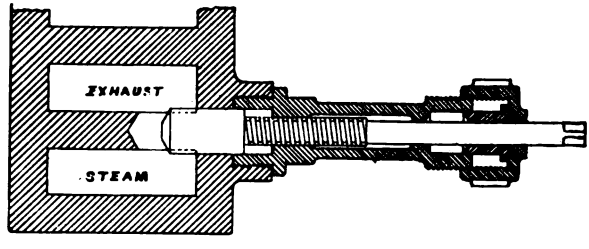


movement with its driving piston, and the other one in an opposite direction, otherwise no reversals at the end of the stroke could occur.

In a direct-acting engine there is no controlling mechanism, such as a crank and connecting-rod, to prevent the piston overrunning its stroke; the arrangement of five ports in the steam-cylinder effectually prevents the piston coming in contact with the heads, and accomplishes it in the following manner: The forward movement of the piston is due to unbalanced pressure; so long as the exhaust is open its forward movement is assured, but when the piston passes over the inner or exhaust port there can be no further escape of exhaust steam, and whatever quantity of steam should be then remaining in the cylinder will be compressed between the tightly-fitting piston and the cylinder-head, forming an elastic cushion and effectually preventing direct contact between the two. It is the practice to allow a considerable clearance in duplex steam-cylinders, for a 12-inch-stroke pump the allowance being usually half an inch at each end before contact; but before this limit is reached the slide-valve belonging to this cylinder will have been carried over to its opposite position, the steam-pressure admitted, and the reversal of the piston movement occurs.

Dash-Relief Valves.— In the larger sizes of cylinders, say 14 inches or more in diameter, a drilled and reamed opening

FIG. 173.

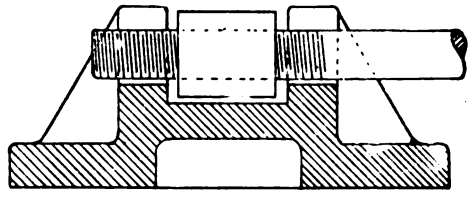


is formed at each end of the steam-cylinder between the steam- and exhaust-ports; this opening is fitted with an

adjustable valve so that a greater or less opening may be secured between these two ports. This arrangement, in one of its forms, is shown in Fig. 173. These are called dash-relief valves, and are intended to regulate the extent of cushioning, adapting it to the peculiarities of the piston movement, and to slightly lengthen the stroke after the piston has closed the exhaust-port, especially when the pump is working slowly and with a heavy load. This communication between the two ports, with its adjustable valve, is simply a controlled leak, useful in lengthening out the stroke in case of excessive cushioning.

Lost motion between each valve and the nut by which it is driven is almost always necessary in the final adjustment of the slide-valve travel, it being a convenient method by which to equalize the length of stroke of the steam-pistons on the two sides of a duplex pump. The amount of lost motion necessary to the proper working of a pump cannot always be determined in advance, but after testing a few pumps from new patterns of any given size, an amount of lost motion necessary to that particular size or pattern of duplex pump is had, and from that point forward the same quantity of lost motion in duplicate pumps will secure substantially the same results as to piston travel. This is not absolute, but it is near enough to manufacture the parts, to assemble the pumps from such patterns and send them to the testing-room for final adjustment. Small pumps, such as boiler-feed and tank pumps, having a stroke of say 9 inches and less, require a lost motion varying from $\frac{1}{8}$ to $\frac{3}{8}$ of an inch. The usual method of construction of the slide-valve and the nut by which it is driven for small pumps is shown in Fig. 174. The valve-nut is simply a square

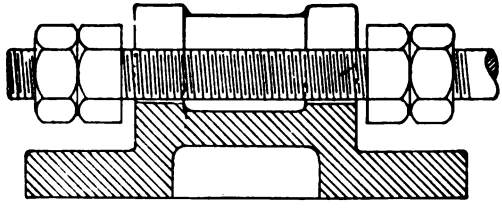
FIG. 174.



block of iron tapped and screwed on the end of the valve-rod as shown. The method of adjustment may be as follows: The steam-pistons are to be placed in the centre of the stroke; the cross-heads are to be fastened to the piston-rods midway of their stroke; the levers connected to the rock-shafts so that the rocker-arms carrying the pins giving motion to the slide-valves are in a vertical line, one rock-shaft pin above the centre, and the other below it; the slide-valves are now to be placed on the valve-seats and centred so as to cover all the ports; the valve-nut must now be centred as shown in the engraving; that is, there must be an equal amount of lost motion on either side between the projections on the valve and the nut by which it is to be driven. This adjustment must be made in both valves in the manner indicated above, after which this detail of the pump is ready for steam.

Duplex pumps of 10 or 12 inches stroke are usually fitted with nuts at each end of the valve, as shown in Fig. 175.

FIG. 175.



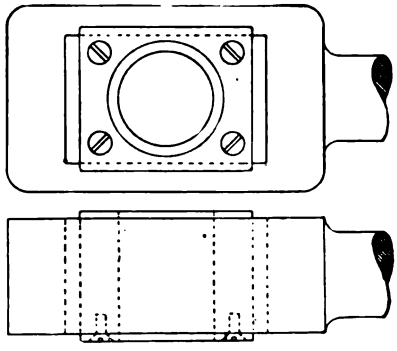
The method of adjustment is the same as described in the preceding paragraph. The nuts offer a greater facility in making adjustments during the operation of testing, and as it sometimes happens that one side of the pump needs an amount of lost motion greater or less than the other, it can be had quickly and without disconnecting the valve-rod from the rock-shaft pin. If care is exercised in tightening the nuts after their adjustment, the probability of their ever working loose is quite remote and rarely ever happens.

Lost-motion links for pumps of 18 to 24 inches stroke are commonly made as shown in Fig. 176. The rock-shaft pin is fitted to the bored hole in the sliding-block. The link

to which the sliding-block is fitted has an end motion suited to the requirements of the pump with which it is to be used.

The amount of lost motion may vary from ½ an inch to 1 inch, depending somewhat on the design of the pump as well as the conditions of service.

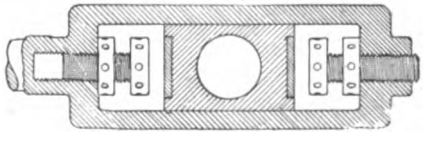
FIG. 176.



When testing a new pump at the works with such a link, a sliding-block much shorter than is likely to be needed is sometimes used, and then by inserting pieces of thin sheet-metal into each end of the link opening, while the pump is in operation, an accurately determined quantity of lost motion can be experimentally secured; then the new sliding-block intended for the pump can be made to an exact length by simply making it to the combined length of the short block and the several thicknesses of sheet-metal used at each end in the slot necessary to give the slide-valves their proper travel.

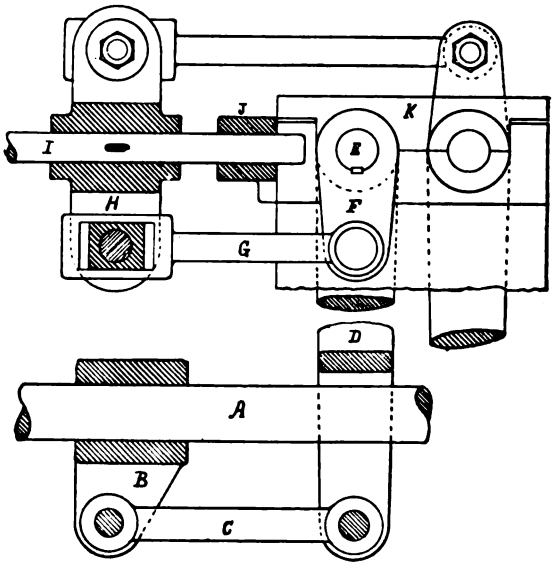
An adjustable lost-motion link, such as shown in Fig. 177, is the kind usually included in the design and construction of large pumping engines. The drawing is nearly self-explanatory; the sliding-block is made with a shallow recess at each end, in which are fitted pieces of raw hide, against which the round heads of the adjusting-screws at each end of the link may touch and communicate motion without noise. The screws are easily adjustable even when the pump is in motion. When the proper amount of lost motion is had, the jam-nuts firmly fix the adjusting-screws in place. This style of lost-motion link permits a very wide range of adjustment.

FIG. 177.



The general features of the valve-gear shown in Fig. 182 are reproduced in Fig. 178 on a somewhat larger scale, and sufficiently sectioned to make the operation more easily understood. *A* is the piston-rod; *B*, a cross-head secured to the piston-rod; *C*, a link connecting the cross-head with the lever *D*; *E*, a rock-shaft operated by *D*, by which is driven the rocker-arm *F*; *G*, a lost-motion link by which the valve-rod cross-head *H* is driven; *I* is the valve-rod by which the slide-

FIG. 178.

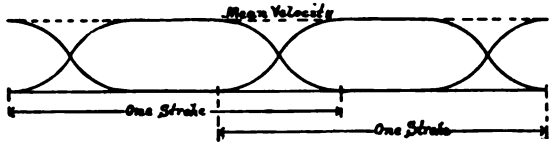


valve is moved; *J* is an outer bearing for the valve-rod; *K* is the upper portion of the cross-stand. This description includes only one set of valve-gear; the duplicate set for operating the other engine is partly shown, and is in all respects the same except one rocker-arm must always be down below the centre of the shaft, and the other one must always be above it; that is to say, one valve-rod must always travel with the piston by which it is driven, while that of the other engine must travel in an opposite direction to that of the piston which

gives it motion. It will doubtless be understood that the piston-rod *A* and the valve-rod *I* are not on the same, but are on opposite, engines. The lever, the lost-motion link, and the valve-rod cross-head, not lettered but shown in the drawing, belong to the opposite engine, and were simply included to show the relation of each to the other.

The action of a duplex valve-movement such as just described must not be confounded with that of two crank pumping engines arranged to work at right angles to each other. It is not the same kind of movement, and the two methods of propulsion have nothing in common. A velocity diagram showing the action of two double-acting pumps with cranks at right angles has already been given in a previous chapter (see Fig. 141). The flow of water from a duplex

FIG. 179.



pump is so entirely different that the illustration, Fig. 179, will hardly be recognized as one performing a similar service, but as a matter of fact it represents approximately the flow from a Worthington pump at each point of the stroke. As soon as one pump begins to slow down at the end of the stroke the other pump starts, so that by combining the flow it will be seen how uniform it is.

In pumping engines of this type the weight of the moving parts is reduced to a minimum, so that the elastic force of the steam practically acts upon the water column directly, the smoothness of working being well illustrated in the above diagram.

The very great success attending the introduction of the duplex pumping engine shows that it well provides the means of pumping heavy columns of water with ease and safety to

the machinery employed, permitting the application of any amount of power required to lift the water column without violent or abrupt action upon the water, thus meeting an acknowledged demand that the rate of movement of the water column through the forcing-main shall be, as nearly as possible, uniform, so that no considerable alteration of pressure shall be shown at any time while the pump is working. It also meets the requirement that the propulsion of the water shall be produced by the use of the smallest practicable amount of moving material for transmitting the force of the steam to the column of water in order to reduce to the lowest point the momentum of moving parts, and the hurtful effects due thereto in case of derangement of the valves or pipes. The time allowed at the end of each stroke before the piston takes up its return motion is sufficient to permit the water-valves to seat quietly, and to allow the incoming supply to completely fill the water-cylinder.

CHAPTER XII.

COMPOUND DIRECT-ACTING STEAM-PUMPS.

THE compounding of direct-acting pumping engines is made necessary if the heat wasted by the exhaust from such engines is to be turned to a useful account by directly assisting in the work of the engine. Single-cylinder direct-acting pumping engines cannot of themselves, as ordinarily constructed, use steam expansively, because the terminal pressure must equal the pump-load, which is, or ought to be, continuous throughout the stroke; no greater steam-pressure is required at the beginning than at its termination. The steam-pressure is, therefore, the same from end to end of the stroke. Compound steam-pumps are now made in considerable quantities, with cylinders adapted to a range of expansion usually from 2 to 4 volumes, depending upon the initial steam-pressure, and whether they are to be operated non-condensing or condensing.

The compounding of steam-cylinders for a direct-acting pumping engine is a different problem to that of an ordinary steam-engine. Practically there is no momentum of moving parts to aid in expansion; there are no fly-wheels or other devices for storing up energy in the early part of the stroke to be given out at the end. The terminal pressure must be sufficiently high to complete the stroke against the water-pressure; the engine is, therefore, working at a disadvantage, and the same degree of economy cannot be expected as if it were an ordinary engine.

Gain by Compounding.—Non condensing compound steam-ends are effective and economical when from 65 to 100 pounds boiler-pressure is used; the gain over the ordinary direct-acting cylinders being from 20 to 35 per cent., depend-

ing on the initial pressure and the ratio of expansion. Non-condensing steam-ends are not recommended for pressures below 50 pounds, as there will not be a saving sufficient to warrant the additional cost of the machinery.

Tandem Direct-Acting Compound Steam-End.—The high-pressure cylinder is supposed to take its steam directly from the boiler and admitted at the same pressure. Practically this is seldom the case, and from 5 to 10 pounds less than boiler-pressure is the ordinary available pressure.

It is not a general practice to supply receivers or tanks with ordinary compound duplex pumps. The high- and low-pressure cylinders are placed tandem to each other, with both pistons attached to the same rod, as shown in Fig. 180. The exhaust from the high-pressure cylinder passing directly into the low-pressure steam-chest, and from thence into the low-pressure cylinder.

The slide-valves of both cylinders have the same movement, the ports being of equal dimensions in the direction of valve travel. The valves having neither lap nor lead, the steam and exhaust are operated at full stroke in both cylinders. The pressure of steam in the low-pressure steam-chest will be variable, depending on the distance which the pistons have travelled towards the end of the stroke.

In this design the low-pressure cylinder is fitted with tie-rods connecting it with the water-end; a substantial cast-iron foot or base under the cylinder affords support for carrying the weight of the entire steam-end, as it is not a common practice to provide the high-pressure cylinder with any means of support other than the end flanges connecting it with the intermediate head. For small pumping engines it is the practice to have the high- and low-pressure valve-seats in the same plane, both valves being driven by the same valve-rod. The lost motion may be had in each steam-chest by a proper adjustment of the nuts at each end of the valve. The double nut shown between the two steam-chests is simply to provide a convenient means of connecting or disconnecting the valve-

rod, should it be necessary at any time to remove the high-pressure cylinder; it has nothing to do with the valve adjustment.

The valve-rod movement is constant, the variation in valve travel being secured by the valve-nuts. The steam- and exhaust-ports must be of the same dimensions in the direction of valve travel, any difference in area that may be required can be had by a proper width of port, as this detail is not affected by the valve travel. The steam-ports in pumping engines are usually two per cent. of the cylinder area, and in compounding these same proportions are used in each cylinder respectively, so that the width of the ports will vary for each diameter. Dash-relief valves are fitted to the low-pressure cylinders only if the engine is to be operated non-condensing, but if condensing, and if the high-pressure cylinder be of a size larger than 12 inches diameter, they may be applied to each end of each cylinder. These valves are not shown in the drawing, but are similar to the one illustrated in Fig. 173. A side-pipe, shown partly in dotted lines and in elevation, connects the high-pressure exhaust-cavity with the low-pressure steam-chest, each side of a duplex engine having its own side-pipe. The steam-pipe shown at the end of the high-pressure steam-chest extends across and connects to the two high-pressure chests, but there must be no connection between low-pressure steam-chests. The exhaust from the low-pressure cylinder may lead directly into the air or into a condenser.

High-Service Attachment.—A compound pump is sometimes required to work temporarily against a water-pressure which cannot be had by the use of the ordinary steam-pressure acting upon the high-pressure piston. Passenger-elevators in office and public buildings are sometimes used to elevate safes and other heavy weights, requiring a water-pressure much greater than is employed regularly. Small water-works pumping engines delivering into a reservoir, stand-pipe, or in direct system, frequently require that the ordinary domestic pressure be doubled for fire-service.

These and other reasons make it desirable that a compound pump be quickly changed to a high-service non-compound when emergencies arise.

To accomplish this requires nothing more than a direct steam connection to the side-pipe (on each side of the pump) leading from the exhaust of the high-pressure cylinder into the steam-chest of the low-pressure cylinder. This pipe must be fitted with a globe-valve for shutting off the high-pressure steam when it is not wanted.

Its action is this: Suppose a non-condensing compound pump to be regularly at work and a fire-alarm be sounded; nothing requires to be done except to open the globe-valve leading from the main steam-pipe into each exhaust side-pipe; by this act the high-pressure cylinders will be thrown out of service, because the exhaust pressure would balance the live-steam pressure and no work could be done. The boiler-pressure being now transformed directly to the low-pressure pistons, which are probably 2½ times the area of the high-pressure cylinder, will so increase the water-pressure that fire-hose connections may be made directly to the fire-plugs. When the fire or other emergency is over this valve may be closed, and the engine goes on with its work as before.

The intermediate head connecting the high- and low-pressure cylinders, as shown in Fig. 180, is one which the writer believes to be superior to all others with which he is acquainted. A loosely-fitting sleeve, preferably of cast iron, is held in place by its flange, and so fitted that it may move easily in any radial direction in the recess between the intermediate casting and the bolted flange back of it. This joint should be well made by scraping to a steam-tight surface on both sides of the flange, otherwise there will be a steam leak between the two cylinders. There may be a reasonable allowance, as shown in the clearance lines, for radial movement of this cast-iron sleeve to allow for any want of original alignment; but as all the work is machine-fitted, there ought not to be any considerable deviation from absolute truth.

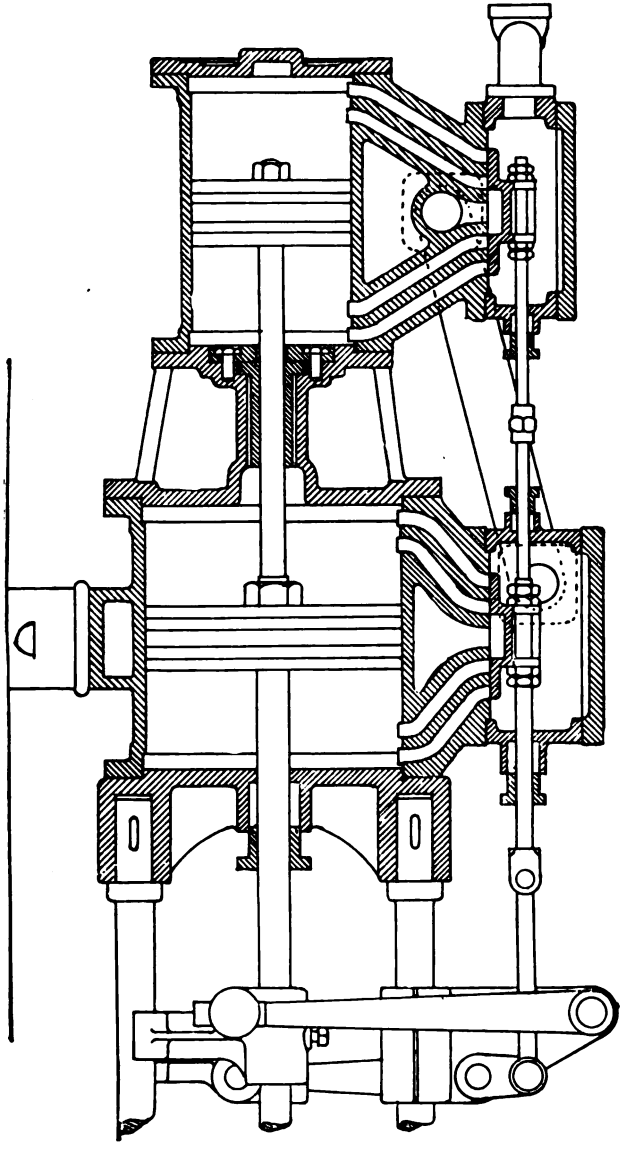


FIG. 180.

The high-pressure piston-rod passing through this sleeve will in a short time, through such lubrication as the cylinder gets, glaze the inside of the sleeve, producing a surface-finish not subject to abrasion, and of great hardness combined with a high polish.

For steam-ends having a stroke of 18 to 24 inches the high-pressure cylinder is sometimes placed between the low-pressure cylinder and the water-end, as shown in Fig. 181. The high-pressure cylinder-head is fitted to receive the tie-bars, but in other respects the design is substantially the same, except that a lost-motion link is included in the valve-gear and no lost motion is given the valves in the steam-chests.

Compound Steam-End for Large Engines. — For pumping engines larger than those described, say from 24 to 48 inches stroke, it is customary to make them with the low-pressure cylinder outside, the high-pressure cylinder inside, as shown in Fig. 182, but without any intermediate head as previously described. In this drawing the usual method of construction is shown, which is to include the flange for bolting the high-pressure cylinder to the low in the same casting with the former, and providing an intermediate cover to the smaller cylinder as shown. The high-pressure cylinder-head through which the piston-rod passes does not receive the tie-bars as in smaller sizes, lugs being included in the cylinder-casting for taking the strain of the engine. Steam-ends of this design have three piston-rods, one for the high- and two for the low-pressure piston,—a detail shown in the half-plan, Fig. 183, in which a cross-head is common to the three piston-rods, as well as the plunger-rod connection leading into the water-end. The low-pressure rods have each a stuffing-box, which is located well forward, almost to the high-pressure cylinder-head, a pipe connection with flange securing each to the low-pressure cylinder.

An advantage which this design offers over the ones previously described is that the high- and low-pressure pistons may be removed, if necessary, without disturbing the main

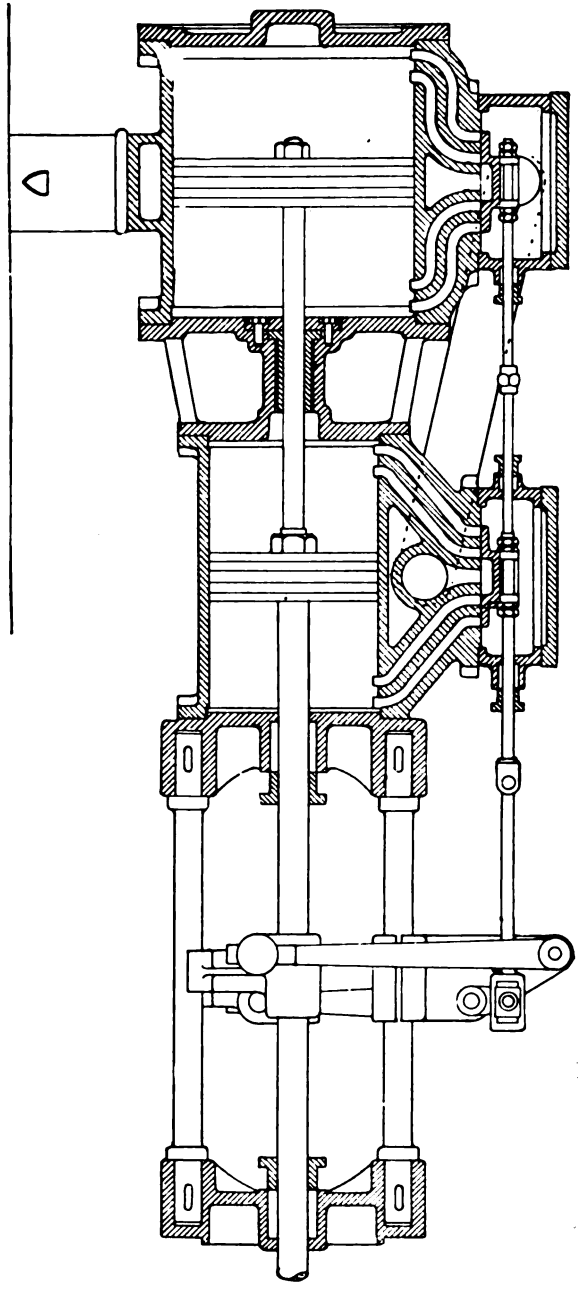


FIG. 181.

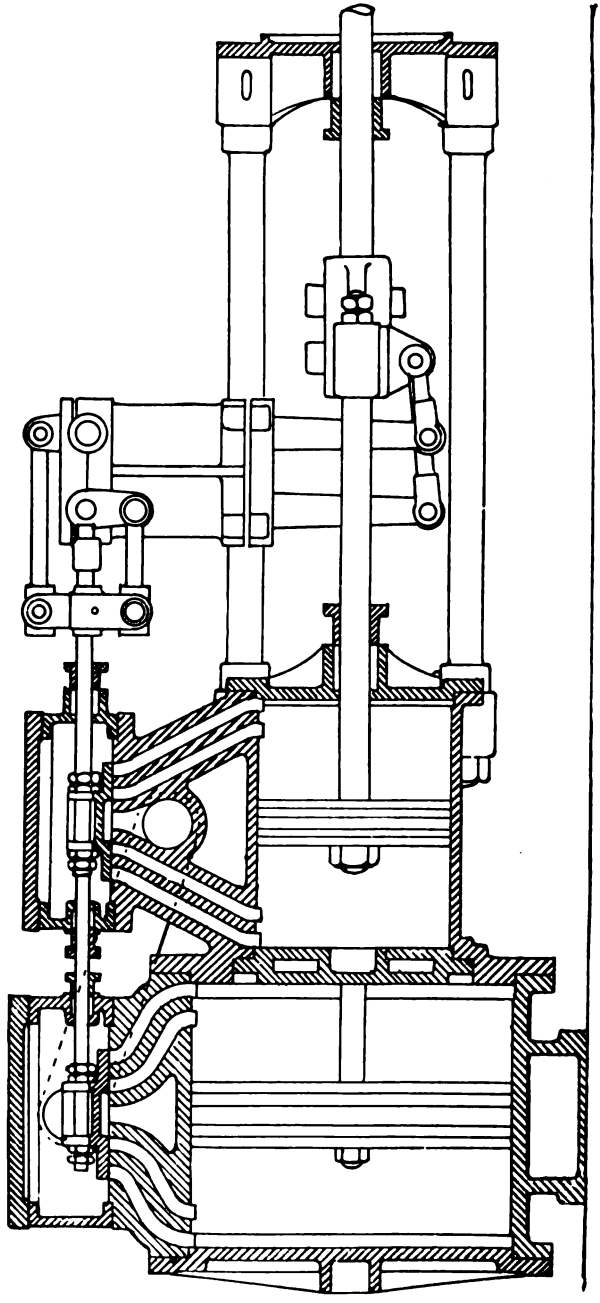


FIG. 182.

portions of the engine. For small engines, the removal of the high-pressure cylinder and intermediate head is not much of an undertaking, but for large water-works engines it would be a serious matter, which the present design makes wholly unnecessary. The valve motion as shown in this engraving has been previously illustrated and described. (See Fig. 178.)

The cross-head for large engines is usually made of forged open-hearth steel; for the smaller sizes a good quality of steel casting has been found to answer the purpose, but cast iron is seldom, if ever, used. The rods should be of a good quality of mild steel; the

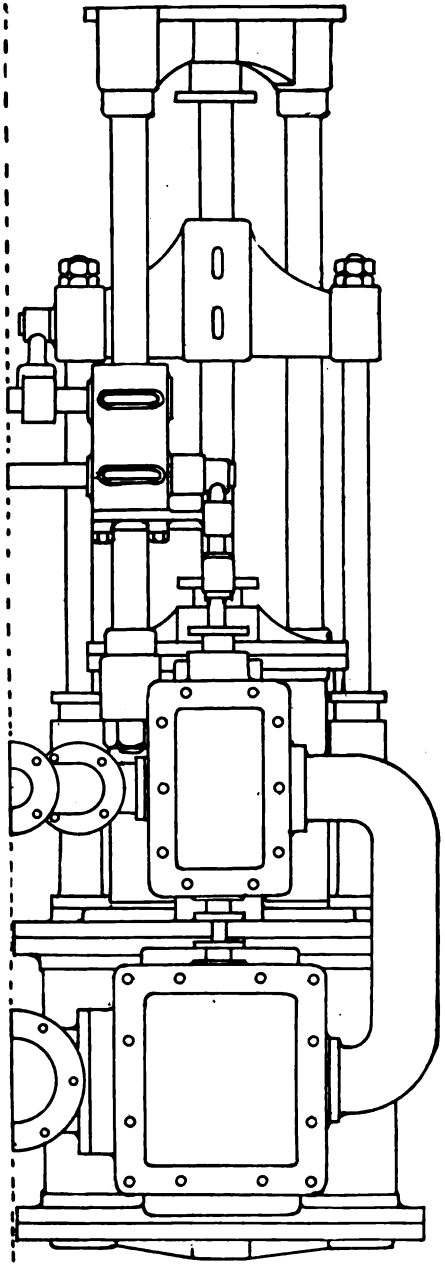


FIG. 183.

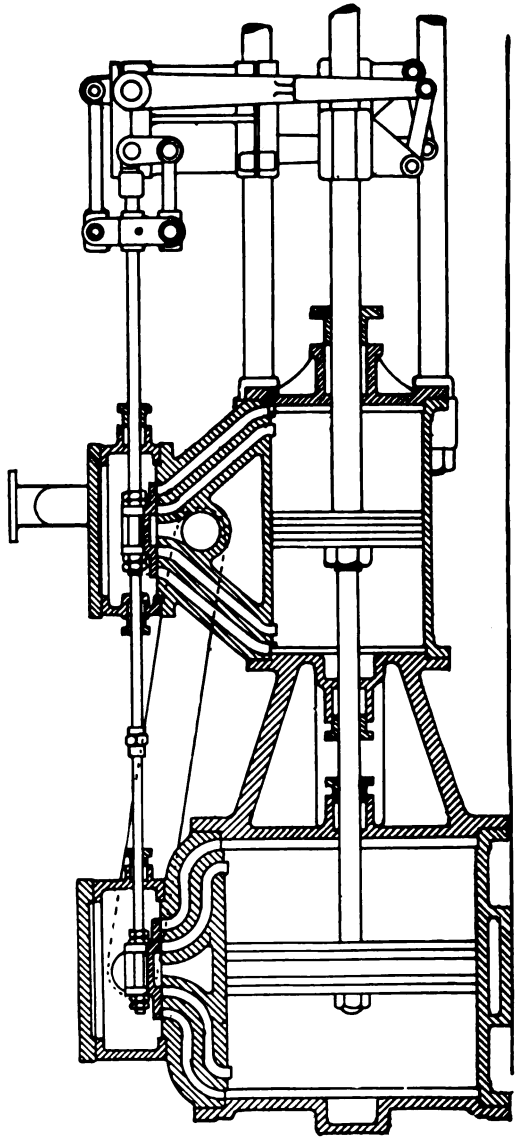


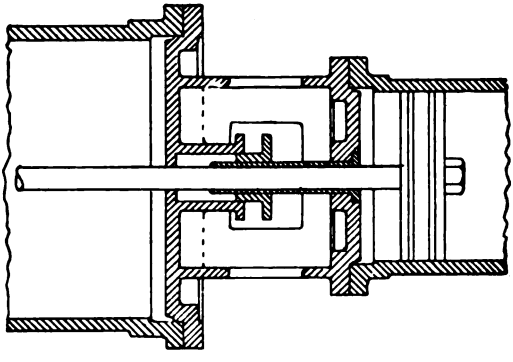
FIG. 184.

writer has used cold-rolled steel in such engines with excellent satisfaction.

Intermediate Head with Stuffing-Boxes.—A compound steam-end with an intermediate head furnished with stuffing-boxes is shown in Fig 184. It does not often occur that such a head must be used to the exclusion of the one previously described, but it has this one advantage, that any leakage around the rod may be detected and remedied by the adjustment of, or the insertion of, new packing. Any one of the several metallic piston-rod packings now offered to the trade may be used in these stuffing-boxes in lieu of the fibrous packing if desired.

Intermediate Head with Protected Rod.—A combination of a cast-iron sleeve with that of a stuffing-box is shown in the sketch, Fig. 185. This is simply a preliminary sketch by the writer, never having been used by him, nor is he sure whether or not it may interfere with another's secured rights. The intent is to protect the high-pressure rod from contact with the atmosphere, by the use of a long sleeve, as well as to prevent leakage from one cylinder to another by means of the stuffing-box and gland shown in the low-pressure head.

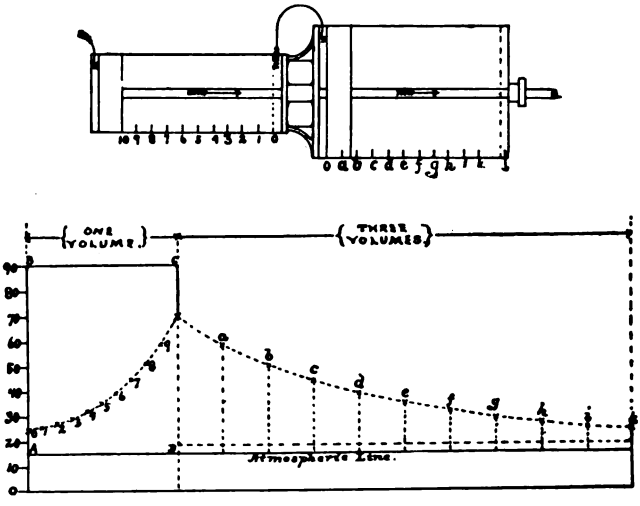
FIG. 185.



Action of Steam in Compound Direct-Acting Engines.—Let the upper part of Fig. 186 represent one side of a tandem compound duplex pumping engine, but in the consid-

eration of the subject it is not necessary to take the duplex feature into account, for the two engines are entirely separate except in the single feature of valve-movement for the distribution of steam, therefore whatever remarks apply to one

FIG. 186.



engine apply equally well to the other. The lower diagram is drawn for 90 pounds absolute pressure; the low-pressure cylinder has an area three times greater than that of the high; both pistons are connected to the same rod, therefore the same length of stroke. This diagram illustrates the action of the steam between the high-pressure piston and that of the low for each one-tenth of one stroke. For convenience we have lengthened the low-pressure cylinder diagram corresponding to three volumes, and we may further suppose the high-pressure cylinder to have a cubic capacity of one foot, or any other volume, real or assumed. The left-hand side of the diagram represents the action of the steam in the high-pressure cylinder, and the right-hand side that of the low. The exact detail of these two cylinders may be the same as the sectional elevation, Fig. 180. No receiver is employed in this engine, but the volume of ports, pipe, and low-pressure steam-

chest is taken to be 50 per cent. of the high-pressure cylinder, the exhaust from the high-pressure cylinder being led directly into the low-pressure steam-chest as shown in the drawing. The engine is represented as non-condensing in the diagram, but as all pressures are reckoned from a vacuum, a slight change in the diagram will make it apply equally well to condensing-engines.

When steam is admitted at *A* the pressure immediately rises to *B*, 90 pounds above vacuum, the piston begins its forward movement, and will continue to the end of the stroke at *C*; the inner edge of the high-pressure piston is marked 10, and the inner edge of the low-pressure piston is marked 0; when, therefore, the one piston has travelled to 9, the other is at *a*; but before considering the expansion curve in the diagram it is necessary to take account of the clearance space between the two cylinders. In engines of this type the steam is exhausted from the high-pressure cylinder at the same pressure used in doing its work, which we have placed at 90 pounds in the diagram. The clearance volume in the ports, side-pipe, and low-pressure steam-chest has been assumed to be 50 per cent. of the high-pressure cylinder. If this space were empty the initial pressure for the low-pressure piston would be: one volume of steam at 90 pounds expanded into 1½ volumes = 60 pounds; but there was remaining in this space from the last stroke a volume of steam having a tension equal to the terminal pressure upon the low-pressure piston at the completion of its stroke, say 24 pounds; this must be taken into account, so that the pressure in the low-pressure chest would be increased thus:

1 vol. at 90 pounds	=	90 pounds	from the H. P. cylinder.
½ vol. at 24 "	=	12 "	terminal pressure in L. P. cylinder.
1.5 volumes	)	102 (68 pounds.	

This 68 pounds becomes, then, the initial pressure for the low-pressure cylinder, as it is also a back pressure against the high-pressure piston. This reduction of pressure occurs at the moment of reversal of the slide-valve in the high-pressure

steam-chest, and at a time when the reversal of pistons is about to occur, but the movement is not yet begun.

Returning now to the consideration of the diagram, Fig. 186, when the high-pressure piston is at 9, and that of the low is at *a*, a reduction of pressure is shown, and it is that due to the expanded volume of steam, or difference between a contraction of one-tenth of the high-pressure cylinder volume and the enlargement by three-tenths in the low-pressure cylinder volume. When the pistons reach 8 and *b* respectively we have the difference between a contraction of two-tenths in the high-pressure cylinder and an enlargement of six-tenths in the low, which reduces the pressure to *b*, and in like manner for other pressures.

It will be a near enough approximation, if we consider the back-pressure line of the high-pressure cylinder, to follow the same curve as the expansion in the low-pressure cylinder, but a slight variation occurs in practice, due to the resistance of the steam flowing through the passages between the two cylinders. The effect of the clearance space between the two cylinders is shown first in the fall of pressure at *c*, in Fig. 186, at which point the highest pressure in the low-pressure cylinder is had and the greatest back pressure in the high-pressure cylinder. The total effect being observable in the difference between the extremes of the curved line terminating at 0.

The exhaust side of the high-pressure cylinder and the pressure side of the low-pressure cylinder are always in direct communication; the pressure must, therefore, be the same in both cylinders. The left-hand diagram shows the back pressure in the high-pressure cylinder until 0 is reached, a similar line is traced in the right-hand diagram until *k* is reached, and the low-pressure exhaust-valve opened to the atmosphere, causing an immediate drop in the pressure to the back-pressure line, which is drawn at 18 pounds, or about three pounds above the atmosphere. The high-pressure cylinder not being in communication with the atmosphere, its terminal pressure is never below that in the low-pressure steam-chest,

a pressure we have estimated to be 24 pounds. The pistons having reached the end of the stroke, the slide-valves moved to the opposite throw, the operation of the engine is now reversed, and as its action is precisely similar to that just described, no further comment is necessary. This tracing of the valve movement of a single pair of compound cylinders is equally applicable to the duplex valve movement, because such steam-cylinders work independently of each other.

Ratios of High- and Low-Pressure Cylinders.—Table XIII. gives the actual dimensions for the several ratios of compounding, but in practice these figures are seldom adhered

TABLE XIII.

SHOWING THE RATIOS OF HIGH- AND LOW-PRESSURE STEAM-CYLINDERS.

The areas are calculated, and opposite each is the corresponding diameter to the nearest ⅛ inch.

High-Pressure Cylinders.		Low-Pressure Cylinders.									
		2 to 1.		2½ to 1.		3 to 1.		3½ to 1.		4 to 1.	
Diameter.	Area.	Nearest Diameter.	Area.	Nearest Diameter.	Area.	Nearest Diamet r.	Area.	Nearest Diameter.	Area.	Nearest Diameter.	Area.
6	28.3	8½—	56.6	9½—	70 7	10⅜ +	84.9	11¼—	99.1	12	113.1
7	38.5	9⅞ +	77.0	11 +	96.2	12⅛ +	115 5	13⅛—	134.8	14	153.9
8	50.3	11¼ +	100.6	12⅝—	125.7	13¼ +	150.9	15 —	176.1	16	201.1
9	63.6	12¾—	127.2	14¼—	159.0	15⅝—	190.8	16⅞—	222 6	18	254.5
10	78.5	14⅛ +	157.0	15⅞—	196.3	17¼ +	235.5	18¾—	274.8	20	314.2
11	95.0	15½ +	190 0	17⅜—	237.5	19 +	285.0	20½ +	332 5	22	380.1
12	113.1	17 —	226.2	19 —	282.8	20¾ +	339.3	22½—	396.0	24	452.4
13	132.7	18⅛ +	265.4	20½ +	331.8	22½ +	398.1	24⅛—	464 5	26	530.9
14	153 9	19¼ +	307.8	22⅛ +	384 8	24¼—	461.7	25⅛ +	538 6	28	615 8
15	176.7	21¼—	353.4	23¼—	441.8	26 —	530.1	28⅛—	618.5	30	706.9
16	201.1	22⅝ +	402.2	25¼ +	502.8	27¾—	603.3	30 —	703.9	32	804.3
17	227.0	24 +	454.0	26⅞ +	567.5	29½—	681.0	31⅞—	794.5	34	907.9
18	254.5	25½—	509.0	28½—	636.3	31⅛ +	763.5	33⅛ +	890.8	36	1017 9
19	283.5	26⅞—	567.0	30 +	708.8	32⅞ +	850.5	35⅝—	992.3	38	1134.1
20	314.2	28¼ +	628.4	31⅝	785 5	34⅝ +	942.6	37⅛ +	1099.7	40	1256.6
21	346.4	29¾—	692.8	33¼—	866.0	36⅛	1039.2	30¼ +	1212.4	42	1385.4
22	380.1	31⅛—	760.2	34¼ +	950.3	38⅛—	1140.3	41⅛ +	1330.4	44	1520.5
23	415.5	32½ +	831.0	36⅛—	1038.8	39⅞—	1246.5	43 +	1454.3	46	1661 9
24	452.4	33⅞ +	904 8	38 —	1131.0	41⅝—	1357 2	44⅞ +	1583.4	48	1809.6

to, as fractional parts of an inch are always avoided in steam-cylinders if whole numbers can be used instead; for example, if a high-pressure cylinder 16 inches in diameter be selected for a given pumping engine, and a ratio of expansion of 3 to 1 was desired, the table gives 27¾ inches as a suitable diameter; the probability is that 28 inches would be selected instead, but the general tendency is towards smaller ratios of expansion. The ordinary or trade sizes of compound steam-ends for engines of 10 to 18 inches stroke are given in Table XIV.

TABLE XIV.

COMMERCIAL SIZES FOR COMPOUND STEAM-ENDS, OR THOSE ORDINARILY FURNISHED DUPLEX PUMPS FROM 10 TO 18 INCHES STROKE.

Diameters.		Areas.		Ratio of Expansion.
High Pressure.	Low Pressure.	High Pressure.	Low Pressure.	
6	10	28.3	78.5	2.78
7	12	38.5	113.1	2.94
8	12	50.3	113.1	2.25
9	14	63.6	153.9	2.42
10	16	78.5	201.0	2.56
12	18	113.1	254.5	2.25
14	20	153.9	314.2	2.04
16	24	201.0	452.4	2.25
18	30	254.5	706.9	2.77

A noticeable feature in this table is the comparatively low ratio of expansion, but it should be borne in mind that these are nearly all small sizes, and are used at pressures rarely exceeding 75 pounds boiler-pressure, and often not more than 60 pounds; the last three or four sizes being, in general, the only ones fitted with condensing apparatus.

TABLE XV.

EXAMPLES OF COMPOUND DUPLEX ENGINE PERFORMANCE.

Size of Engine.	Water-Pressure.					Steam-Pressure.			Exhaust Back Pressure.	Size of Pipes.				Static Head in Feet.
		At Speed in Strokes per Minute.								Water.		Steam.		
	At Rest.	Pressure.	Speed.	Pressure.	Speed.	Boiler.	H. P. Cylinder.	L. P. Cylinder.		Suction.	Delivery.	Steam.	Exhaust.	
8 and 12 × 7 × 10 . .	48	50	106	52	200	60	56	14	2	6	5	2	4	111
8 and 12 × 7 × 10 . .	60	69	100	72	200	55	50	13	2	5	5	2	3	118
8 and 12 × 8 × 10 . .	38	52	150	57	180	70	63	18	3	6	3½	2	2	88
8 and 12 × 8½ × 10 .	29	37	160	. .	. .	65	53	13	0	6	4	2	2½	67
10 and 16 × 10¼ × 10	36	44	160	47	200	72	60	17	1	6	5	2	3	83
10 and 16 × 10¼ × 10	52	52	110	56	160	60	57	15	1	7	7	2	4	120
12 and 18½ × 12 × 10	52	54	100	57	180	70	60	15	3½	7	6	2	3	120
12 and 18½ × 12 × 10	62	65	100	70	160	60	55	15	3	7	6	2	3	143
18½ and 29 × 16 × 18	68	75	60	80	92	75	47	9	0	14	12	4	9	157
21 and 36⅜ × 6½ × 36	200	112/5	60	. .	. .	100	94	25	3	6	4	5	8	462

The last pump tabulated was performing the very unusual service of pumping through a delivery 24 miles long.

The others represent fair average performance of compound duplex non-condensing pumping engines.

Steam-Jackets.—It is not a common practice to jacket the smaller sizes of steam-cylinders, whether in single or duplex pumping engines; that is to say, for sizes smaller than 18- and 30-inch compound. It is not believed, for the smaller sizes at least, that the future saving will repay the original cost; but if a partial jacketing be thought desirable, it should be the low-pressure cylinders rather than the high. If the high-pressure cylinder only is jacketed, the exhaust from it into the low-pressure cylinder suffers by excessive condensation because of its lower temperature; but if the latter cylinder is jacketed, this condensation is less in quantity, and in so far a step towards more economical working.

The usual practice is to jacket the barrel of the cylinder only; the heads are seldom jacketed except in very large engines. A steam-jacket must be designed with reference to proper circulation and drainage, otherwise it may prove

worthless, and defeat the very end to which it was applied. The steam used in the jacket should be at full boiler-pressure, and provision made for returning any water of condensation back to the boiler.

The degree of expansion in ordinary pumping engines is not sufficiently great to require the many refinements which are thought to be necessary in the case of large engines where high pressures are employed and expansion carried to the extreme limit; in the latter engines jackets are always employed.

The actual value of the steam-jacket, first applied by Watt, has always been called in question. Tredgold, the earliest writer on the steam-engine, condemned it, but it steadily grew into favor with engineers, and it is now the general belief that expansive engines cannot work to any good purpose without one. A few years ago it was thought that its application effected a saving from 10 to 30 per cent., but the lowest of these two figures is not now believed to be possible even under the best conditions.

The best series of experiments undertaken up to this time to determine the exact truth in relation to steam-jackets were those undertaken and conducted by Professor J. E. Denton in 1889, the high-duty Corliss engine of the Pawtucket water-works having been placed at his disposal.

The details of this engine are tolerably well known, but it may be well to repeat that it is a horizontal cross compound engine, steam-cylinders 15 and 30⅛ inches bore; water-cylinders, 10.52 inches; stroke of all pistons, 30 inches; clearance, high-pressure cylinder, 4 per cent.; low, 3.7 per cent. Diameter of rods, 2¾ inches. Ratio of volumes of cylinders, 4.085. Average cut-off in high-pressure cylinders, one-fourth, and in low, one-third. Jackets envelop the barrels but not the heads of both cylinders, and steam of full boiler-pressure is used in each. The heads are not jacketed, but contain passages leading to and from the ports. The condensed steam from the jackets is pumped into the feed-pipe at a point between the boiler and hot-well.

The experiments were conducted along the lines of every-day usage and service, and have an especial value for this reason. The influence of the steam-jackets on the Pawtucket pumping engine form the subject of an elaborate and convincing paper by Professor Denton, published in Vol. XI., "Transactions of the American Society of Mechanical Engineers." The conclusions reached by him are as follows:

1. That the averages of results of indicator-cards, taken in the most careful manner with the best modern indicators, show a possible saving from the use of jackets amounting to from 0.13 to 0.35 pound of steam per hour per horse-power, but that these amounts are within the limit of error to which the determination of indicated horse-power and cut-offs are subjected, so that

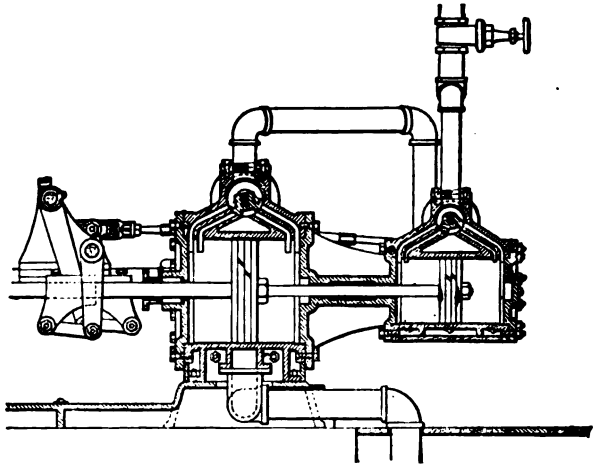
2. The most that can be claimed for the jacket is that it probably caused no loss, and may possibly cause a saving not exceeding 3 per cent. of the total steam consumption.

Lagging steam-cylinders with wood or other non-conducting material in lieu of jackets is very common. This does not, of course, take the place of a steam-jacket, nor is it the intention to arrest cylinder condensation by its application; its only function is to prevent as far as practicable external radiation from the steam-cylinder. The best non-conductors of heat are hair felt, cork, fossil meal, magnesia, charcoal, and rice chaff. A cylinder clothed with hair felt, and afterwards covered with reeded black walnut staves held in place by a polished brass band about two inches wide at each end of the cylinder, makes a good and at the same time a very neat finish for a steam-end.

Circular slide-valves are sometimes used, as shown in Fig. 187. The arrangement of the ports and of the valve does not essentially differ from that of the ordinary duplex steam-cylinder, except in the matter of change of form of the valve and its seat from flat to circular. The necessary lost motion for the slide-valves is secured by the employment of

adjustable lost-motion links at the rocker-arms; the method of operating the valve within the steam-chest being similar to that of a Corliss steam-valve makes it inconvenient to have any lost motion there. The centring of the two valves is secured through the use of a right- and left-hand nut attached to the valve connecting-rod, by which it may be lengthened or shortened to get any suitable distance from centre to centre.

FIG. 187.



This nut is shown between the two cylinders. It is not necessary that the two valve-seats be bored to the same diameter, but care must be exercised in designing that the valve-movements be precisely alike in the matter of admission and port closure: this is easily secured by making the high-pressure ports shorter, or by so proportioning the high-pressure valve-lever that the two valves shall open and close exactly alike.

A tank-engine is a duplex pumping engine in which there is one high-pressure cylinder and one low-pressure cylinder placed side by side as in the construction of an ordinary

duplex pump, with a large receiver or tank, into which the high-pressure cylinder exhausts, and from which the low-pressure cylinder draws its supply. This type of engine probably originated with Ernest Wolff,* who patented it in England in 1834. The essential features of his patent were thus described by him:

"The invention consists of the combination of two or more engines, each complete in all its parts, and so disposed that while the first receives steam at one, two, or more atmospheres of pressure, the next engine is moved by the steam that escapes from the first. In the last engine the steam is condensed in the ordinary way, or escapes in the atmosphere. The work supplied by the several engines is applied to the same shaft, or to several combined, or to independent shafts.

"It is sometimes useful to have between the cylinders an intermediate reservoir to regulate the pressure; this may be placed with advantage at the base of the chimney, so as to maintain or raise the temperature and the pressure of the steam in its passage from one cylinder to the other. Indeed, if necessary, the heat may be supplied by a special fire-box.

"It is often necessary to employ a special pipe with a stop-cock to admit the steam from the boiler to an intermediate reservoir in order to give the machine the power of starting any crank. This direct introduction may be employed to increase for a time the power of the engine."

Certain modifications of this engine were patented by Worthington in 1871, and engines for situations favorable to its use have been built from time to time, but its introduction has not been general. In these engines a large receiver or tank is a necessity, not less than say ten times the volume of the high-pressure cylinder; it is from this tank that the low-pressure cylinder draws its supply as if from a steam-boiler. This large tank is necessary that the contribution of the high-

* This invention must not be confounded with another engine patented in 1804 by Woolf. Notice the difference in spelling.

pressure cylinder to its contents, or the withdrawal of steam from it by the low-pressure cylinder, is not sufficient to vary its pressure in any considerable or troublesome degree. The condensing apparatus is attached, of course, only to the low-pressure cylinder. A careful consideration of the working conditions under which this engine is found will show that it must be accurately proportioned as well to its work as to the proposed pressure of steam, in order to secure anything like an equal division of the resistance between the two pistons. And it is necessary to do this with some reasonable degree of exactness, as upon it depend that uniformity of water-flow and stability of pressure which give to this engine its advantages as an hydraulic motor. Economically this engine holds about the same position as the four-cylinder engine, and costs somewhat less. It is not, however, universally applicable, and is not likely ever to supersede the present types of duplex, either ordinary or compound.

Power of Compound Pumping Engines.—The paper on the "Power of Compound Pumping Engines," by John W. Hill, in *Engineering News*, vol. xxi., is laid under contribution for the subjoined tables and formulas.

Table XVI. contains the general data for the steam-ends of four proportions or classes of compound, duplex, direct-acting pumping engines; thus,—

Class A embraces engines where the diameter of the low-pressure piston is 1.5 times the diameter of the high-pressure piston, as 8 and 12, 12 and 18, 16 and 24, etc.

Class B embraces engines where the diameter of the low-pressure piston is 1.6 times the diameter of the high-pressure piston, as 10 and 16, 15 and 24, etc.

Class C embraces engines where the diameter of the low-pressure piston is 1.57 times the diameter of the high-pressure piston, as 14 and 22, 21 and 33, etc.

Class D embraces engines where the diameter of the low-pressure piston is two times the diameter of the high-pressure piston, as 12 and 24, 16 and 32, 18 and 36, etc.

The table contains the piston displacements, clearances, and expansions, based upon volume (1) and expansion (1) in high-pressure cylinders, the hyperbolic logarithms for the effective expansions, and the reciprocals of total expansion.

TABLE XVI.

COMPOUND DUPLEX DIRECT-ACTING ENGINES.

Cylinder Volumes and Clearances stated in Terms of High-Pressure Cylinders and Steam Full-Stroke, High-Pressure Cylinders.

Terms.	Class of Engine.			
	A.	B.	C.	D.
Diameter C. P. cylinder (H. P. = 1.00) .	1.5	1.6	1.57	2.0
High-pressure cylinder, area	1.000	1.000	1.000	1.000
" " clearance	0.0630	0.0630	0.0630	0.0600
Intermediate space	0.3388	0.3388	0.3388	1.1200
Low-pressure cylinder, area	2.2500	2.5600	2.4700	4.0000
" " clearance	0.1125	0.1280	0.1235	0.2040
Expansion, high-pressure cylinder	1.0000	1.000	1 0000	1.0000
" intermediate space	1.3190	1.4391	1.4350	2.2500
Expansion, low-pressure cylinder	1.8254	2.0200	1 9640	2.2584
Total expansion	2.6000	2 9067	2.8182	5.0814
Effective "	1.8254	2.0200	1.9640	2.2584
Hyperbolic Logarithm	0.6018	0.7031	0.6750	0.8146
Reciprocal total expansion	0.3846	0.3440	0.3548	0 1968

Table XVII. is calculated for compound duplex direct-acting pumping engines, when worked *condensing*, and gives for each of the four classes the mean effective pressure reduced to work of low-pressure piston for the absolute pressure of column 1; for example:

The steam-end of a compound duplex direct-acting condensing engine has cylinders 16 and 24 inches diameter, and the steam-pressure is 70 pounds. What size or diameter of pump will this work against a pressure or head of 80 pounds (184.64 feet)?

The data will be found in Class A.

TABLE XVII.

MEAN EFFECTIVE PRESSURES, ENGINE WORKED CONDENSING.

Compound Duplex Direct-Acting Pumping Engines; Steam Full-Stroke, High-Pressure Cylinders; Counter-Pressure, 3.5 Pounds Absolute.

Initial Pressure Absolute.	Class of Engine.			
	A.	B.	C.	D.
	Constants.			
	0 72869	0.68252	0.69542	0.46587
65	43.864	40.864	41.702	26.781
70	47.508	44.277	45.679	29.111
75	51.152	47.689	48.656	31.440
80	54.795	51.102	52.133	33.769
85	58.438	54.515	55.611	36.098
90	62.082	57.927	59.088	38.428
95	62.725	61.340	62.565	40.659
100	69.369	64.753	66.042	43.087
105	73.012	68.165	69.519	45.418
110	76.656	71.578	72.996	47.746
115	80.299	74.990	76.473	50.077
120	83.943	78.403	79 950	52.407
125	87.586	81.816	83.427	54.736
130	91.229	85.228	86.904	57.063
135	94.873	88.641	90.382	59.392

The area of the low-pressure piston (24 inches diameter) is 452.39 square inches, and mean effective pressure for absolute pressure of 85 pounds (70 by gauge) is 58.438, and moment of water-load—

$$452.39 \times 58.438 \times .85 = 22471.24 \text{ pounds.}$$

The head is 80 pounds, and area of plunger which a steam-end containing high-pressure piston 16 inches diameter, and low-pressure piston 24 inches diameter, with steam-pressure of 70 pounds, will drive—

$$\frac{22471.24}{80} = 280.89 \text{ square inches,}$$

corresponding to a diameter of 19 inches nearly. Two such pumps, double-acting, will deliver at 100 feet piston-speed per minute 4,034,309.9 gallons per day of 24 hours.

Table XVIII. contains the mean effective pressures for compound duplex direct-acting pumping engines worked *non-condensing*.

TABLE XVIII.

MEAN EFFECTIVE PRESSURES, ENGINE WORKED NON-CONDENSING.

Compound Duplex Direct-Acting Pumping Engines; Steam Full-Stroke, High-Pressure Cylinders; Counter-Pressure, 17 Pounds Absolute.

Initial Pressure Absolute.	Class of Engine.			
	A.	B.	C.	D.
	Constants.			
	0.72869	0.68252	0.69542	0.46587
65	30.364	27.364	28.202	13.281
70	34.008	30.777	31.679	15.611
75	37.652	34.189	35.156	17.946
80	41.295	37.602	38.633	20.269
85	44.938	41.015	42.111	22.599
90	48.582	44.427	45.588	24.925
95	52.225	47.840	49.065	27.159
100	55.869	51.253	52.542	29.587
105	59.512	54.665	56.019	31.978
110	63.156	57.078	59.496	34.246
115	66.799	61.490	62.973	36.577
120	70.443	64.903	66.450	38.907
125	74.086	68.316	69.927	41.236
130	77.729	71.728	73 404	43.563
135	81.373	75.141	76.881	45.892

Tangyes's Compound Pump.—The Floyd and Morton direct-acting pumping engine, owned by Tangyes, Soho, England, is illustrated in sectional elevation in Fig. 188, which is a reproduction of the drawing attached to their American patent, and is thus described by them:

"The high-pressure steam-cylinder and its appliances may be of any ordinary construction suited for the application of this invention,—such, for instance, as that used in the pump known as 'Tangyes's Special Pump,'—and we will describe, with reference to the accompanying drawing (which is a longitudinal

vertical section), the application of the invention to a pump of that kind, from which its application to pumps of other descriptions will also be understood.

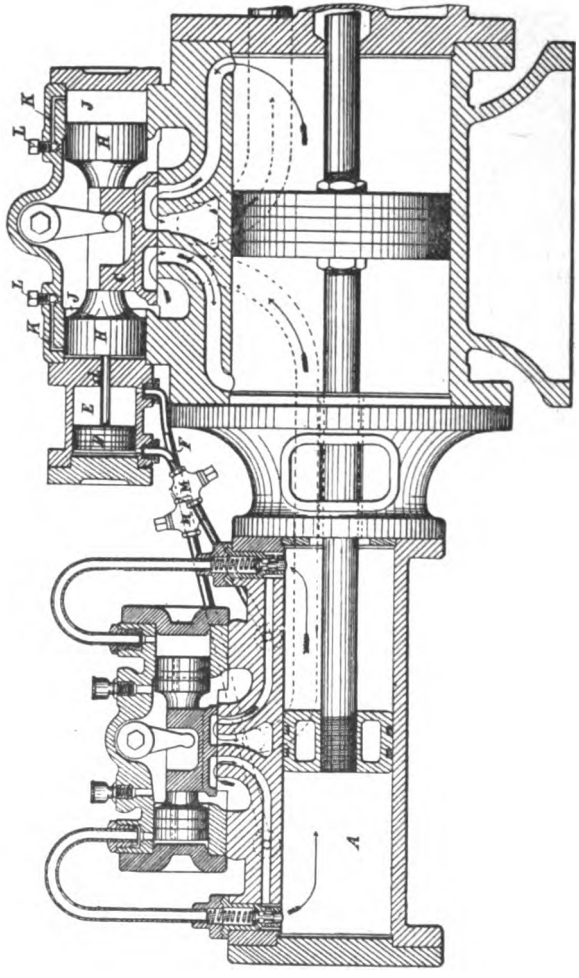


FIG. 188.

"*A* is the high-pressure cylinder and *B* is the low-pressure cylinder. The slide-valve *C* of the low-pressure cylinder *B* is not mechanically connected with, but is independent of, the valve of the high-pressure cylinder, and is operated by a

piston, *D*, working in an auxiliary steam-cylinder, *E*, preferably in line with the valve *C* of the said low-pressure cylinder *B*, which may receive its steam from the high-pressure cylinder in the usual or any convenient way. The ends of the said auxiliary steam-cylinder *E* are connected by means of pipes, *F* and *G*, or equivalent steam-ways to the steam-ports or ends of the high-pressure steam-cylinder *A*, as shown. The slide-valve *C* in the steam-chest of the low-pressure steam-cylinder *B* may be of the D or other type, and the said slide-valve is fitted with piston-valves, *H H*, the pistons of which work in short cylinders, *J J*, at opposite ends of the steam-chest, and the piston *D* is connected to one of the said piston-valves, *H*, by the rod *d*.

"In the arrangement illustrated we use the piston-valves *H H* in the steam-chest of the low-pressure cylinder *B* for controlling the motion of the piston *D* of the auxiliary cylinder *E* in the following manner: Each end of the steam-chest where the pistons *H H* of the slide-valve *C* work is provided with an auxiliary port, *K*, which auxiliary ports connect the ends of the steam-chest *J J* in which the slide-valve pistons *H H* work with the interior of the steam-chest in which the slide-valve *C* works. The auxiliary ports *K K* described may be capable of being opened and closed and regulated by means of screwed plugs or valves, *L L*. By means of the said auxiliary ports *K K* in the steam-chest the space between the back of each of the piston-valves *H H* of the slide-valve *C* and the end of the steam-chest *J J* in which it works is charged with steam, and when the piston *D* of the auxiliary cylinder *E* is moved by the steam entering its cylinder by the pipes *F G*, or steam-ways in connection with the high-pressure cylinder *A*, the steam is compressed in one or the other of the spaces *J J* at the ends of the low-pressure steam-chest, so as thereby to cushion the slide-valve piston and retard the motion of the auxiliary piston *D*. By these means the main pistons in the high- and low-pressure cylinders *A* and *B* are caused to make a pause at the ends of their strokes, thereby permitting the pump-valves to seat themselves quietly on their seats.

"We do not limit ourselves to this arrangement for controlling or retarding the action of the auxiliary piston *D*, as in place thereof an oil-, water-, or air-cylinder may be used, the displacement of the oil, water, or air in the said cylinder by the motion of the piston effecting the retardation of the slide-valve *C*, as hereinbefore described; or the retardation of the motion of the low-pressure slide-valve *C* may be effected by regulating the supply of steam to the auxiliary cylinder by placing stop-valves, *M M*, in the pipes *F G*, by which the said auxiliary cylinder *E* is connected with the steam-ports or ends of the high-pressure steam-cylinder.

"In a compound pump or pumping engine with more than two cylinders the low-pressure cylinder aforesaid would become an intermediate cylinder having its valve actuated by steam, as described, and the slide-valve of the third cylinder would be connected with the ports or ends of the intermediate cylinder and be operated by steam, as described, with respect to the low-pressure cylinder of a two-cylinder compound pumping engine. In the same manner the invention may be applied to a fourth cylinder, if desired."

Compound Direct-Acting Engine with Isochronal Valve-Gear.—The isochronal valve-gear by the Gordon Steam-Pump Company, illustrated in Fig. 168, is further shown in its adaptation to a compound engine in Fig. 189. The working of the valve-gear in the compound engine is in no respect different to that already described, excepting, of course, in that due to its different application.

In this design the condenser and air-pump are placed in a pit between the steam and water-ends. The necessary motion for operating the air-pumps is obtained through a vibrating beam receiving its motion from a suitable cross-head attached to the main piston-rod; brackets are bolted to the under side of the girders joining the steam- and water-ends, in which are included the bearings for the vibrating-shaft. From this same vibrating-shaft motion is given the sliding-cylinder by means of connecting-rods, the centres of which are indicated by

Fig. 189.

Digitized by Google

dotted lines; the movement of the sliding-cylinder and its influence upon the auxiliary cylinder is the same as described on page 218.

The illustration, Fig. 189, represents a single compound engine, but this valve-gear permits the placing of two engines side by side, and so arranged that they may work together as a pair, or independently as single engines, if necessary.

The folding plate, Fig. 190, gives an illustration of a pair of the same machines arranged to work independent of each other as independent isochronal engines, and also provided with duplex gear, by which they can, at will, be run as a compound duplex engine. In this construction all of the advantages of the varying systems are combined, and while the machine may be run as a duplex machine it is also capable of running independently, so as to permit one side to do its work while the other side is at rest or undergoing repairs.

Davidson Direct-Acting Triple-Expansion Pumping Engine.—This is a horizontal direct-acting steam-pump, with the three steam-cylinders in line with the water-cylinder. Fig. 191 is an elevation and plan of the engine. The intermediate cylinder is placed nearest the water-cylinder, and then come the low-pressure and the high-pressure cylinders. Steam is led by pipes from each cylinder to the next. The valves for the high-pressure and the low-pressure cylinders are plain slide-valves, worked by direct connections with the main valve on the intermediate cylinder, which is similar in design to the valve used by Mr. Davidson on compound and single-cylinder pumps. (See Fig. 164)

With a view to saving space, the intermediate and the low-pressure cylinders are set close together, the rear end of the first forming the head of the second. This made it impossible to use a central piston-rod between these cylinders, there being no room for a stuffing-box. The low-pressure piston, therefore, has two piston-rods, which pass outside the intermediate cylinder, but underneath its lagging, and are connected to a cross-head fastened to the main piston-rod. The stuffing-boxes

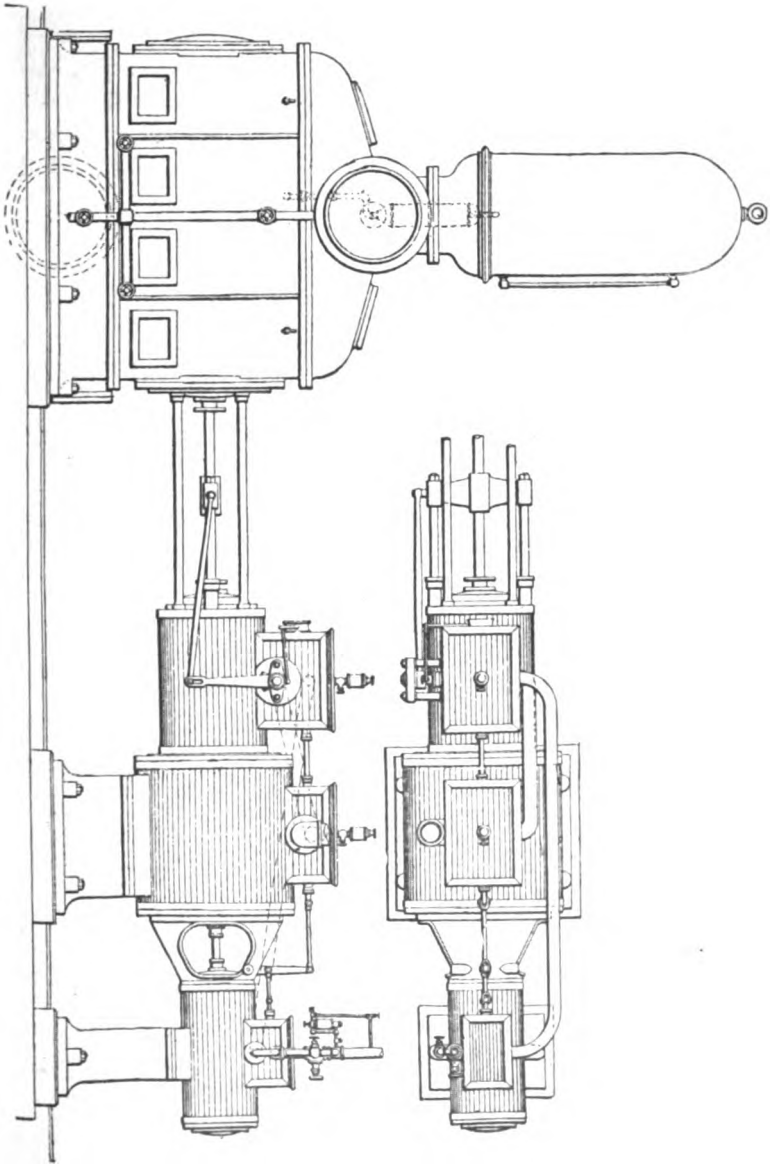


FIG. 191.

for these rods are placed on the end of the intermediate cylinder farthest from the low-pressure cylinder, and the rods work through brass bushings, thus making it as easy to care for these stuffing-boxes as for those in the ordinary position. To facilitate starting the engine, a by-pass leads from the main steam-pipe to the pipe connecting the high-pressure and intermediate cylinders, by which both these cylinders can be worked with live steam until the engine is under way.

A sectional elevation of the steam-end is shown in Fig. 192. The valve-cylinder attached to cylinder A is 12 inches in diameter, giving a large area for wear, and steam-ports of ample size. The area of the valve-piston is 113 square inches, so that the steam-pressure furnishes ample force to work the valves of the three cylinders.

The low-pressure cylinder and steam-chest are shown at B. The steam-ports are 1¼ by 14 inches. The high-pressure cylinder, shown at C, has its left head cast in one piece with the right head of the low-pressure cylinder, and the pistons of the two cylinders are connected by a central piston-rod. The steam-ports for this cylinder are ½ by 5¼ inches.

The steam-cylinders are 11½ inches, 20 inches, and 36 inches diameter, with a common stroke of 36 inches. They are made of hard cast iron and jacketed with live steam, the condensation being returned to the boiler by a small independent pump. Steam-pistons are in one piece, with packing-rings sprung in, and re-enforced by brass springs pressing outwards. The cylinders are covered with approved non-conductor, handsomely lagged with walnut strips, held in place by finished brass bands.

The water-cylinder is of cast iron, 36 inches diameter by 36 inches stroke, with removable lining of hard cast brass and brass valve-seats, guards, and springs. It is arranged for suction connection on either side, as convenient, and has ample openings for removal or examination of valves. The capacity of the cylinder is such that at 45 strokes per minute a delivery of 7000 gallons per minute is obtained.

The water-piston is packed with square fibrous packing for

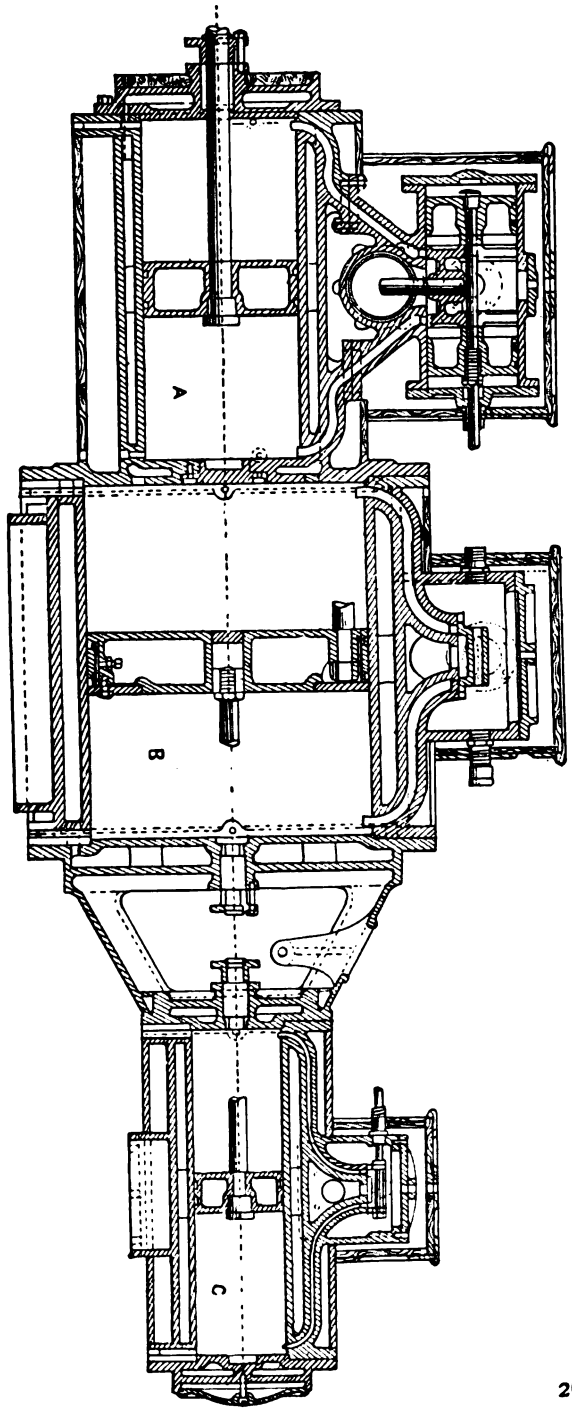


Fig. 192.

taking wear and with cupped leather rings for preventing leakage.

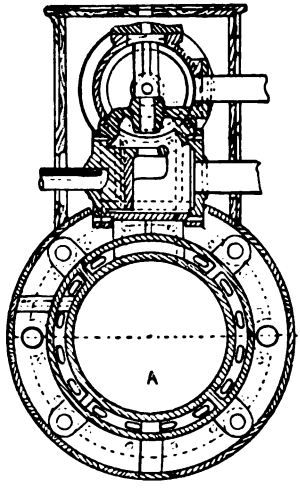
The air-vessels are of cast iron, with brass water-gauges and air-charging device whereby air is automatically maintained at proper height.

The piston-rods are of machinery steel, secured to the pistons with brass nuts and jam-nuts and to the cross-head by steel keys. The tie-rods for uniting the steam- and water-cylinders are of iron, neatly turned and finished.

Oil-cups are placed at all points where lubrication is required, and a brass one-quart sight-feed lubricator is placed on the high-pressure steam-cylinder.

With each engine is furnished a Davidson independent condenser, with steam-cylinder 8 inches and air-cylinder 14 inches in diameter and a stroke of 16 inches. The air-cylinder is lined with brass, and has brass valve-seats and guards and sheet-metal valves. Condensers have vacuum-breaking attachments to insure against flooding the steam-cylinder of main engines and possible damage to them by reason of inattention or negligence on the part of those in charge.

FIG. 193.



Cross-section through valve-cylinder.

It is of interest to compare this pumping engine with other well-known pumping engines. Of course the first question which arises is as to comparative duty or fuel economy. In a power-plant working under the conditions present in most water-works pumping-stations, the matter of fuel economy is rightly considered of the first importance. A high-duty pumping engine, very expensive in first cost, may be much more economical in the long run than a cheap engine giving only moderate fuel economy.

So far no direct-acting steam-pump (with the exception of the Worthington engine with the high-duty attachment) has been able to approach in fuel economy the fly-wheel engines, which store up the force exerted by the steam during the first part of the stroke and give it out again on the last part of the stroke when the steam is expanded and is pressing with less force upon the piston.

As to the question how the pump illustrated may be expected to compare in duty with the best types of compound duplex pumping engines, it is difficult to form any opinion without a comparison of indicator diagrams. The duplex compound can cut off the stroke and secure considerable expansion of steam in the first cylinder, and has the great advantage of securing a steady movement of the water at the pump end by the mutual action of the two cylinders.

The triple-expansion engine here illustrated has the advantage of carrying on the expansion in three cylinders. If we assume that in this engine the valves of the three cylinders open and close in exact unison, and each valve remains open for the whole stroke, which we believe is practically correct, then the number of expansions obtained is the ratio between the areas of the high-pressure and the low-pressure pistons. Since these are to each other as the square of their diameters, or as $11\frac{1}{2}^2 : 36^2$, the number of expansions will be $1296 \div 132.25 = 10$, nearly. While the highest duties recorded by pumping engines have been obtained with greater ratios of expansion than the above, many excellent records have been made with about this number of expansions. The pumping engine at Lawrence, Mass., designed by E. D. Leavitt, expands the steam about 22 times, and the Pawtucket engine, designed by Geo. H. Corliss, expands the steam about 18 times. The ordinary duplex direct-acting engines have, as a rule, much smaller ratios of expansions.

The position of the high-pressure cylinder in this engine seems rather unfortunate for the attainment of a high duty, on account of the long pipe necessary to carry the steam to the intermediate cylinder. The clearance of the intermediate

cylinder is practically increased by the volume of this pipe. This arrangement of the steam-cylinders was adopted to permit the examination and packing or readjustment of each of the three steam-pistons with the least delay and inconvenience.

The piston-speed when working at the contract capacity is 135 feet per minute. This again is much lower than that of the fly-wheel engines. The Lawrence engine works at about 216 feet piston-speed, and the Pawtucket engine at about 250 feet per minute. The speed of 135 feet per minute, however, is certainly high for direct-acting pumps, and we understand that the engine is capable of as high speeds even as 200 feet, or 67 strokes per minute. It is remarkable that smooth motion and freedom from shock can be obtained from so large a pump moving its own valves.

It is to be hoped that a thorough duty test may be made of these engines after they are in place, that it may be known what economy can be attained with engines of this class.

CHAPTER XIII.

FIRE-PUMPS.

THE Underwriter Pump is the name adopted by The Associated Factory Mutual Insurance Companies to designate a steam fire-pump built in strict accordance with certain specifications, the salient points of which are given below, taken from a special circular prepared for the use of their agents and patrons. The writer regards this circular as containing the most intelligent and carefully-prepared specifications for steam fire-pumps that have come to his knowledge.

INTRODUCTION.

Fire-pumps at the factories which we insure are so very frequently found incapable of being started promptly when tried for our inspectors, particularly at the spring inspection, and are found in so very many cases to be incapable of delivering anywhere near their alleged or rated capacity without violent "hammering," as to make it plainly evident that some improvement in fire-pumps is greatly needed at the average mill in one or all three of the following particulars:

The construction of the pump.

The erection and fitting up of the pump.

The care of the pump.

Steam fire-pumps are ordinarily sold, rated, and expected to run at double the speed of a pump for boiler-feeding or any ordinary water-supply; they do not ordinarily receive such care or attention as a steam-engine or other machine on which the product of the mill depends. Being constantly subject to dampness, warmth, and contact with condensed steam, their parts are especially subject to rust. After weeks

and months of disuse a fire-pump should be capable of being instantly started and run at full speed by a man who is excited and perhaps unskilful.

Therefore a fire-pump needs more strength, better workmanship, and better protection against rust than does a pump for boiler-feeding or ordinary miscellaneous use; and the high speed at which it is expected to run demands the large steam valve-ports and large suction-valve areas specified below.

These specifications cover only certain general features of the design, and may be viewed as an effort to indicate to those whom we insure the kind and quality of pump which we believe is needed.

Beyond these features which are specified, the various pump-makers are free to follow each his own judgment and experience in details of shop practice, to the end that competition in excellence may be encouraged. On the pump-maker rests the responsibility for first-class workmanship, material, and strength of castings, bolting, etc.

SPECIFICATIONS.

ARTICLE 1. Only "duplex pumps" are acceptable for steam fire-pumps.

(So-called "duplex" pumps, consisting of a pair of pumps with "steam-thrown valves" actuated by supplemental pistons, are not acceptable.)

The experience of our inspectors goes to show that duplex pumps are more certain of starting after long disuse. The whole power of the main cylinder is available for moving a corroded valve or valve-rod, whereas on a single pump with a steam-thrown valve no such surplus of power is available.

Further, the direct-acting duplex has the great advantage over a fly-wheel pump of not suffering breakage if water gets into the steam-cylinder.

SIZE OF PUMP.

ART. 2. The present multiplicity of odd sizes is confusing, and different makers estimate the capacity in gallons according to different arbitrary standards.

(To simplify matters we recommend that only the four different sizes given below be recognized or considered.)

Capacity to furnish Standard Fire-Streams.	Capacity in Gallons per Minute delivered at Full Speed.	Ratio of Piston Areas.	(See Article 4 for commercial sizes.)
One stream.	250 to 320	4 to 1	*Too Small for a Fire-Pump* except as an auxiliary. Convenient for regular use in extensive boiler-feeding or domestic supply. At speed suitable for continuous use delivery would be but about half this, or about 150 gallons per minute.
Two streams.	500	4 to 1	*Ordinary Size for Small Mills.* This needs steam-boilers of at least 100 horse-power capacity to drive pump at rated full speed and maintain 100 pounds water-pressure. A steam-pressure of at least 40 pounds is needed *at the pump* to give this water pressure of 100 pounds at full speed. (The foregoing is for pumps in the best of order.)
Three streams.	750	3 to 1	*Ordinary Size for General Use.* This needs steam-boilers of at least 115 horse-power and a steam-pressure of at least 48 pounds square inch *at the pump* to drive at full speed and at 100 pounds square inch water-pressure. Pressure at boiler must be a little more to allow for loss of steam-pressure between boiler and pump. For cases where boiler-pressure is unusually low (less than 50 pounds), use a pump with larger (4 to 1) steam-cylinder.
Four streams.	1000	3 to 1	*Size for Large Factories.* Requires 150 horse power boiler capacity and at least 44 pounds square inch steam-pressure. (The boiler horse-powers above are reckoned on the A. S. M. E. basis of 30 pounds of water evaporated or consumed per I. H. P. per hour, and are for pumps in the best order, and with parts as given in these specifications.) (Pumps in poor order or too tightly packed will require more steam than stated above.)

Where more than 1000 gallons capacity is needed it is generally best to provide two pumps.

Two hundred and fifty gallons per minute is our standard allowance for a good 1⅛-inch (smooth-nozzle) fire-stream.

(From 15 to 20 automatic sprinklers may be reckoned as discharging about the same quantity as a 1⅛-inch hose-stream under the ordinary practical conditions as to pipes supplying sprinkler and hose systems respectively.)

It is not expected to provide pump capacity sufficient to supply all the sprinklers in a large room for operation all at once; it being assumed that if controllable by sprinklers the fire will be controlled before spreading outside the area covered by from 25 to 75 heads.

CAPACITY, HOW COMPUTED.

ART. 3. The capacity of a pump depends on the speed at which it can be run.

To compare different pumps, some definite speed must be agreed on as a basis.

We have adopted,—

For direct acting steam-pumps of } 10-inch stroke,—Full speed = 75 revolutions per minute.

For direct-acting steam-pumps of } 12-inch stroke,—Full speed = 70 revolutions per minute.

The rated capacity is, therefore, to be computed as the product of the speed just stated, by the plunger displacement for one revolution, corrected for piston-rod volume and for full nominal length of stroke, and with 10 per cent. deduction for slip and loss of action.

(One revolution means one complete circuit of the motion of any of the reciprocating parts of the pump, and for a duplex pump is equivalent to four single strokes. "Revolutions" is a term less liable to be misunderstood than "strokes.")

It is all right to run fire-pumps at the highest speed that is possible without causing violent jar or hammering within the cylinders.

Although a 12-inch stroke is 20 per cent. longer than a 10-inch stroke, the maximum delivery of a 12-inch-stroke pump will not be so much as 20 per cent. more than the delivery of a 10-inch-stroke pump of same diameter, for the reason that, having to travel farther, it cannot make quite so many revolutions per minute before jar will begin.

On the other hand, it will deliver somewhat more, perhaps 12 per cent. more, than the 10-inch-stroke pump, because it can get equal delivery with a less number of reversals of motion per minute.

COMMERCIAL SIZES.

Number of Standard 1⅛-Inch Fire-Streams. (Each 250 Gallons per Minute.)	Ratio of Areas Steam to Water-Piston.	Nominal Capacity. Gallons per Minute Full Speed.	Nearest Commercial Size. 10-Inch Stroke. Diam. Steam-Cylinder. Inches.	Diam. Water-Cylinder. Inches.	Nominal Length of Stroke. Inches.	Exact Capacity Underwriters' Rating. Gallons per Minute. (Including 10 per cent. deduction for slip, etc.)	Nearest Commercial Size. 12-Inch Stroke. Diam. Steam-Cylinder. Inches.	Diam. Water-Cylinder. Inches.	Nominal Length of Stroke. Inches.	Exact Capacity Underwriters' Rating. (Including 10 per cent. deduction for slip, etc.)
These two pumps being so small as to be seldom demanded for fire-pumps, makers are hardly expected to prepare special patterns for them at present.		Too small for a fire-pump alone.	10 × 5 × 10			220	10 × 5 × 12			246
			As between these two pumps, a somewhat greater price for the second is reasonable on account of its 12 per cent. greater capacity.							
	One stream.	4 to 1	320	12 × 6 × 10		321	12 × 6 × 12			359
			As between these two pumps just given above, the second rates at 12 per cent. greater capacity than the first, and a price 12 per cent. greater would therefore not be unreasonable.							
Two streams.	4 to 1	500	16 × 8 × 10			569	14 × 7 × 12			484
			As between these two pumps just given above, the second rates at 15 per cent. less capacity than the first, therefore the second should be sold about so much cheaper.							
Three streams.	3 to 1	750	16 × 9¼ × 10			767	16 × 9 × 12			807
			As between these two pumps just given above, the second rates at 5 per cent. greater capacity than the first.							
Four streams.	3 to 1	1000	18½ × 10½ × 10			989	18 × 10 × 12			1007
			As between these two pumps last mentioned, the second rates at 2 per cent. greater than the first.							

No pump with areas less than 3 to 1 will, except under special circumstances, be accepted as a fire-pump.

It has been common to make all fire-pumps with water-plunger of only one-fourth the area of steam-piston, with the idea that the pump could thereby be more readily run at night, when steam was low.

The capacity in gallons is thus reduced 25 per cent. as compared with a 3 to 1 plunger on the same steam-cylinders.

Often, especially with large pumps, this is a mistake, for though the pump might start and give a few puffs, it withdraws so much steam that it can run effectively for but an extremely short time without the boiler fires are first aroused to make fresh steam to replace that withdrawn.

A steam-piston relatively larger than necessary is a source of weakness. It takes more volume of steam, and gives more power with which to burst something if the throttle is opened wide suddenly during excitement.

ART. 5. The inside-plunger pattern of pump is preferred to the water-piston pattern for all situations where water is reasonably free from grit or mud, and will generally be best for all New England situations.

For pumps in the West or South, or for water like Ohio river water, a water-piston with packing or a packed plunger gives much better wear.

By means of the priming-tank and pipes hereinafter specified, the plunger pattern is enabled to "get hold of its water" as quickly as the piston pattern even under extreme lifts, and our experience goes to show that the plunger pattern is much better adapted to start free after long disuse or unskilful treatment.

SIZE-PLATE.

ART. 6. Every steam fire-pump must bear a conspicuous statement of its capacity, thus:

> 16 × 9¼ × 10.
> Capacity 750 gallons per minute, or three good 1⅛-inch smooth nozzle streams.
> Full speed, 75 revolutions per minute.
> For fire purposes never let steam get below 50 pounds, nights or Sundays.

This plate to be of porcelain-like enamel, bearing black

letters ½ inch high on a white ground. The letters being of enamel and burned in.

The plate is to be securely attached to the inboard side of the air-chamber.

BRASS FITTING.

ART. 7. Fire-pumps are to be brass fitted throughout,—viz., both piston-rods for their whole length are to be preferably of solid rods of special bronze, equal in quality to Tobin bronze, but rods of steel covered with a brass shell ⅛ inch thick throughout both water- and steam-ends are acceptable.

(Common foundry bronze composition is not dense or strong enough for this purpose.)

Valve-rods are to be preferably of solid Tobin bronze, or may be brass covered through their bearings in stuffing-boxes.

Water-plungers are to be of solid brass or bronze, and the ring in which they slide is also to be of brass or bronze. The composition of the plunger and its ring should be of very dissimilar alloys to insure good wearing qualities.

All six stuffing-boxes are to be bushed at the bottom with a brass ring with suitable neck and flange, and the follower or gland is to be either of solid brass, or is to be lined with a brass shell 3-16 inch thick, united with a flange covering the end next the packing.

STRENGTH OF PARTS.

ART. 8. The maker is to be understood to warrant each pump built under these specifications, to be at time of delivery, in all its parts, strong enough to admit of closing all valves on water outlet pipes while steam-valve is wide open and steam-pressure 80 pounds, and agree to test it before shipment from his works.

(In other words, although these pumps are not expected to be designed for a regular working water-pressure of 240 or 320 pounds, it is expected that bolts, shells, rods, etc., will be figured to stand this comparatively quiet, temporary, high pressure exclusive of further allowance for initial strain due setting up of bolts, with a factor of safety of at least four.)

(This high test pressure is analogous to the custom of proving all common cast-iron water-pipes to 300 pounds and all common lap-welded steam-pipe to 500 pounds per square inch, even though it is to be regularly used at much less pressure.)

We are assured that castings no heavier than at present used by the best makers will stand this test, *if properly shaped and liberally bolted.*

ART. 9. The pump is to be warranted so designed, with such arrangement of thickness of metal that it shall be safe to instantly turn a full head of steam on to a cold pump without cracking or breaking the same by unequal expansion, due to this cause.

SUCTION-VALVES.

ART. 10. The net areas of ports in suction-valves for each end of each plunger is to be not less than 50 per cent. of area of plunger, where normal stroke is 10 inches, and for pumps of 12-inch stroke, valve area is to be 56 per cent. of plunger area.

(Experience indicates that it is preferable to secure this area by a larger number of 3-, 3½-, or 4-inch valves rather than by increasing diameter of valve.

The valves probably rarely lift so high in operation that area of cylinder under their edge half equals the area of ports; hence, strictly speaking, it is probably a question of securing the greatest convenient development of total valve circumference rather than merely large port area.)

FORCE-VALVES.

ART. 11. The force-valves need have but two-thirds the total port area specified for the suction-valves.

SPRINGS.

ART. 12. All valve-springs to be coiled on a cylindrical arbor (conical springs are not approved). The spring to be properly guided and held from drifting sidewise out of centre by ends resting in grooves, and a suitable metal plate to be interposed between the rubber valve and the spring.

Valve-spring and valve-spindle or stud are to permit an extreme lift equal to one-fifth diameter of rubber valve, and are to be so formed by screwing in on a taper or otherwise, that this amount of free lift cannot be restricted.

Stiffness of springs is for the present not specified, but left to the discretion of each maker. Preferably one rather stiff standard spring had best be uniformly used for all force-valves, while for the suction-valves three grades of stiffness can be used with advantage.

Thus the standard will serve for all lifts up to 18 feet.

An extra mild spring can be used for higher lifts, and an extra stiff spring where pump takes its suction under 5 pounds or more pressure may aid pump in running smoothly at high speed.

VALVE-SEATS.

ART. 13. Valve-seats to be all of U. S. gun-metal composition and firmly secured into valve-deck by forcing in on a screw-threaded taper, or by forcing in on a smooth taper and expanding out its lower edge below the valve-deck.

STICKING OF VALVES.

ART. 14. A serious objectionable feature of fire-pump valves, and one for which no satisfactory remedy has been put in practice as yet, is that if the pump is left standing or unused for several weeks or months the rubber valve-disks are sometimes liable to become stuck to their seats, and if suction has a high lift there may not be vacuum enough to tear all the suction-valves open.

At present we can only repeatedly urge that steam fire-pumps be started enough to limber them up, at least once a fortnight.

Perhaps some non-corrosive alloy of nickel or aluminum can be found which rubber vulcanized with sulphur would not corrode and adhere to, and experience in other arts indicates that rubber will not adhere to a block-tin face even with extreme long standing. This is a subject well worth investigation by pump-makers.

STEAM-PORTS.

ART. 15. The minimum area of each exhaust-steam passage, at its smallest section, is to be not less than 4 per cent. of the area of the piston from which it leads.

This is a large increase over the size heretofore common, but indicator-cards which we have taken from pumps of several different makes indicate this to be one of the points in which improvement is most needed to accommodate the high speeds at which fire-pumps are always supposed to run, and this unrestricted exhaust aids very materially in giving steadiness to the jet of water.

The admission-ports may be not less than 2½ per cent. of piston area.

Although from structural reasons the steam-port has usually heretofore been made of same size as exhaust-port, it can perhaps be advantageously made shorter.

STEAM-CLEARANCE SPACE.

The clearance space between face of piston and cylinder-head must be reduced to smallest possible amount, and these contacting surfaces be flat, without projections or recesses, excepting the one for the piston-rod nut

Some makers, with the idea that a fire-pump need not be economical, have not taken pains to keep these waste spaces small.

Securing small clearance costs almost nothing but care in design, and is often of value, since at many factories boiler capacity is scanty for the large quantity of steam taken by a fire-pump of proper size.

The edges of the steam valve-ports and likewise of the valve are to be accurately milled, or chipped and exactly filed to templets, true to line, and the face of valve and its seat accurately scraped to a plane surface, all in a most thorough and workmanlike manner and equal to high-grade steam-engine work.

STEAM SLIDE-VALVE ADJUSTMENTS.

(Adjustable valves are not recommended.)

It is recognized that the practice of making adjustable valve-tappets located outside the steam-chest is a good thing on a large pump in constant service and operated by a skilled engine-driver, but for the infrequently used ordinary fire-pump the utmost simplicity is desirable, and it is best not to tempt the ordinary man in charge to "readjust" the valve-gear.

Since the number of different sizes of pumps called for in these specifications are few, it is recommended that gauges and templets be prepared by which the valve position can be set once for all without necessity or ready means for re-adjustments.

CUSHION-VALVES.

ART. 16. Cushion-valves regulating the amount of steam-cushion at ends of stroke, by controlling a passage communicating between steam- and exhaust-ports, are recommended for the 750-gallon and 1000-gallon pumps. They may, if the maker thinks it desirable for simplicity, be omitted on the pumps of 500 gallons per minute or less capacity.

In order to obviate the possibility of the pump pounding itself to pieces in case of a sudden release of load, as by a break in suction- or delivery-mains, we recom-

mend that the steam-cushion release be through independent ports terminating ½ inch back from cylinder-head, instead of through the induction-port opening, as at present common.

This makes the pump safer in case cushion-valves are unskilfully left open too wide.

STROKE GAUGE.

ART. 17. A length-of-stroke index or gauge will be insisted on in all cases. These are to be of satisfactory strong and simple form for at all times rendering obvious the exact length of stroke which both pistons are making, and thus calling attention to improper adjustment of cushion-valves or stuffing-boxes.

PIPE SIZES.

ART. 18. Suction- and discharge-pipe connections must have standard flanges to connect with pipes of the sizes given below.

Size of Pump. Gallons per Minute.	Diameter of Suction-Pipe. Inches.	Diameter of Discharge-Pipe. Inches.	Steam-Pipe.	Exhaust Pipe.
320	6	5	2½	3
500	8	6	3	4
750	10	7	3½	4
1000	12	8	4	5

These suction-pipe sizes, although larger than heretofore common, are believed to be amply justified by our experience, and exert a powerful influence towards enabling the pump to run smoothly at high speed with water-cylinders filling perfectly at each stroke. No defect is more common than restricted suction-pipes.

There should preferably be three suction entrances,—viz., one each side and one at end of pump. One of these openings is designed for the attachment of the suction air-chamber, and the providing of two others is often a great convenience in adapting pump to any particular location, or in permitting drafting from two different sources of supply.

A single central end suction-opening is, however, acceptable.

VACUUM-CHAMBER.

ART. 19. The suction air-chamber (often called a vacuum-chamber), or its equivalent mentioned below, is to regularly form a part of every pump made under these specifications.

This may, however, be omitted *under special agreement* for cases where suction is from an open reservoir by a pipe of full specified size less than twenty feet in total length, but flanges must be arranged so it can be readily added afterwards if found necessary.

If so desired, the vacuum-chamber may consist of a closed vertical pipe of same diameter as suction, and 6 feet long.

This vacuum-chamber must be attached to the pump in the most direct way practicable, but provision must be made for attaching it in such manner as not to prevent readily taking off the cylinder-heads.

For pumps taking feed under a head, a vertical open pipe of same diameter as suction, reaching up to five feet above level of feed-water, is much preferable to a closed suction-chamber, since the latter is liable to become filled with water. If a great head or other cause makes this impracticable, the vacuum chamber should be so connected and fitted with gates that it may readily have water emptied and be filled with air while pump is in motion.

AIR-CHAMBER.

A large air-chamber is more necessary on a fire pump than on other pumps; without it the hose vibrates so that holes are quickly worn through.

An air-chamber of hammered copper, and warranted tested under a hydraulic pressure of not less than 300 pounds per square inch, is preferable to cast iron, as, holding the air better and being lighter, it wrenches and strains the pump less when running fast and shaking.

The air-chamber will be acceptable, however, if made of cast iron; but then it is to be warranted subjected to a hydraulic test of 400 pounds per square inch before connection to pump, and is to be thoroughly painted inside and out to diminish its porosity.

The form of air-chamber combined with the elbow beneath it, and also the relief-valve and hose-valve connections, should all be carefully designed to *make the whole height the very least possible.* Keeping this weight low makes the pump run steadier and brings less strain on the flanges at high speeds.

SIZE OF VACUUM- AND AIR-CHAMBERS.

	Vacuum-Chamber is to contain:	Air-Chamber is to contain:
320-gallon pump.	8 gallons.	10 gallons.
500- " "	13 "	17 "
750- " "	18 "	25 "
1000- " "	24 "	30 "

These volumes are based on making suction-chamber for long pipes six times, and pressure-chamber eight times one displacement of one plunger.

Where the suction will never be from a pipe under pressure, and where the suction-pipe has a total length of less than 25 feet, vacuum-chambers of from half to two-thirds the above will be acceptable.

PRESSURE-GAUGES.

ART. 20. A water-pressure gauge provided with ¼-inch cock with lever handle is to be provided with the pump, and connected close to the air-chamber.

The gauge itself is to be of the 5-inch iron case duplex spring pattern (such as used on locomotives), this kind of gauge being believed to be the best for withstanding the vibrations which cause gauges on fire-pumps so often to be unreliable.

A steam pressure-gauge of the same kind is to be attached to the steam-chest inside the throttle-valve.

SAFETY-VALVE.

ART. 21. A safety- or relief-valve of the Ashton, Crosby, or other similar and accepted pattern is to be regularly included in the price, and is to be attached to each pump; preferably extending horizontally inboard from base of air-chamber, so that its hand-wheel for regulating pressure is within easy reach.

This valve is to be set ordinarily at a working pressure of 100 pounds to the square inch, and is to be of such capacity that it can discharge the full throw of the pump at ⅔ speed, at a pressure not exceeding 125 pounds per square inch. It is to be provided with a hand-wheel, *marked very conspicuously* to show direction of turning, thus:—OPEN ➡

It should not have a locked adjustment, and does not need a device for lifting valve from seat by hand.

Until further experiment it will be assumed that—

For 320-gallon pump a 2½-inch Ashton valve (or equivalent) is suitable.
" 500- " " 3- " " " "
" 750- " " 3½- " " " "
" 1000- " " 4- " " " "

This should discharge by a vertical, downward pipe, about 2½ feet long, opening into a cone or tunnel, fitted to a 6-, 8-, or 10-inch waste-pipe.

This very short open part at the cone being for rendering it obvious to pump operator whether water is wasting through relief-valve.

This cone and its pipes are not included in price of pump.

DRIP-COCKS.

ART. 22. Three-eighths-inch brass drip-cocks with a lever-handle are to be provided on both ends of each water-cylinder and each steam-cylinder, for effectually draining the same. A similar drain-cock is also to be provided for drawing off the water above the upper valve-deck.

A ¼-inch air-cock with lever-handle is to be attached to the cover over water-cylinders.

(Cocks with lever-handle are used on account of showing to every passer-by whether they are open or shut.)

PRIMING-PIPES.

ART. 23. Each pump is to be regularly fitted with 1-inch brass priming-pipes before leaving the shop, as follows (these being included in its price), beginning at a 2 x 1 x 1-inch brass tee close to pump beneath delivery-flange, and thence extending to four ¾-inch valves, one of which leads into each of the four plunger-chambers. These priming-pipes should not connect into suction-pipe, lest priming-water be thereby wasted, and should not connect with chamber above force-valves.

For cases where pump only takes its suction under a head, these priming-pipes may be omitted, but openings for them into the pump-shell must be provided and fitted with screw-plugs.

A priming-tank (not included in pump contract unless at an extra charge) will in all cases be a requirement in the instructions for properly setting up a fire-pump to which the water does not flow under a head. This tank must be situated with its bottom not lower than 5 feet above the pump, and containing a volume of water equal to half the nominal capacity of the pump in gallons per minute, said priming-tank to be for the sole and exclusive use of the fire-pump, and connected with nothing else whatever.

Having this provision for priming quickly and surely, a foot-valve at bottom of suction-pipe is not needed.

ART. 24. Each pump is to be regularly fitted, not as an extra, but as a part included in the cost of the pump, with hose connections, consisting of Chapman Valve Company's 2½-inch Straight-way Hose Valves (not including cap and chain), attached to the neck of air-chamber. The number of these is to be as follows:

For the	300-gallon pump	One.
	500- "	Two.
	750- "	Three.
	1000- "	Four.

The screw-head at end of these valves for connecting to hose is to be either fitted to a hose-coupling furnished by the customer, or left with the thread uncut.

ART. 25. Tests for acceptance after pump is set up at its destination:

1. Pump being set up in good order and properly packed, is to run smoothly without slamming or jumping or hammering, at its full rated speed (of 70 or 75 revolutions per minute), and make full normal stroke, meanwhile maintaining a water-pressure of 100 pounds per square inch while furnished with

45	pounds	square inch	steam-pressure	for the	500 gallons	(4 to 1).
50	"	"	"	"	750 "	(3 to 1).
45	"	"	"	"	1000 "	(3 to 1).

Water-pressure being measured at base of air-chamber.
Steam-pressure being measured close to steam-chest.

Start slow. Gradually and alternately open throttle to give speed required, and if hose is short or discharge too free, close or adjust outlet-valve to give back pressure enough to raise water-gauge to about 100 pounds.

Pump is to run at full speed free from thumping without necessity for "snifting" air into suction.

2. During this trial it is preferable to discharge the water through two, three, or four 1⅛-inch smooth nozzles (according to the number the pump is rated to supply, as per Art. 4). Each nozzle being connected to the hose outlet on the pump by 150 feet of 2½-inch rubber-lined hose, and meanwhile note the pressure of water.

The 100 pounds water-pressure is intended to allow for friction in hose lines 150 feet long. If shorter lines are used, the same nozzle delivery will be obtained with less pressure at the pump.

3. While thus playing through the nozzles the hose near the pump should lie quiet, or with but little sliding to and fro on the ground near the pump. One of the nicest points about building a pump is to so design its valves and gear that the pulsations of the pump will not quickly wear holes in the hose by causing it to rub back and forth at each beat of the piston, and the quietness with which the hose lies is a good index of the pump-maker's skill in securing uniform delivery.

4. An experiment may then be made to determine the greatest speed at which pump can be run, at 100 pounds water-pressure, before beginning to jar or pound seriously, cushion-valves meanwhile being adjusted.

This may be tried by adding another line of hose at some hydrant near, and seeing how well the pump will fill this extra nozzle in addition to those it is rated to deliver.

Count the revolutions, and note the steam- and water-pressures meanwhile.

5. With all water outlets closed and with sufficient steam admitted to give 80 pounds average water-pressure, the internal leakage is to be so small that pump will not make more than one revolution per minute.

The safety-valve should be on during this test, lest pressure be accidentally carried too high.

6. With all water outlets *nearly, but not completely*, closed and safety-valve screwed down and steam admitted sufficient to give a water-pressure of 240 pounds per square inch, pump moving very slowly meanwhile, all joints about the pump are to remain substantially tight.

7. With all other outlets closed and the safety-valve adjusted to 100 pounds, then on starting the pump this safety-valve should be sufficient to discharge the full delivery of the pump at fifty revolutions per minute, with a pressure of not over 125 pounds.

8. A brief experiment may then be made, with cushion-valves wide open, to determine greatest speed at which pump can be run in this condition without injurious hammering.

FINALLY, COST.

ART. 26. A pump built as specified above will cost more than a fire-pump of same size built as heretofore, but we believe it well worth the extra cost; and the tests on the few pumps already made indicate that the first cost per gallon actual capacity is, by reason of smooth running and higher speed, practicable, even less than for the ordinary style.

The main points of difference are:

Pump has brass plungers instead of cast-iron ones.

Pump has bronze piston-rods and valve-rods instead of iron.

Pump has brass-lined stuffing-boxes instead of iron.

Area of water-valves 25 to 50 per cent. greater.

Steam- and exhaust-passages 20 to 50 per cent. greater.

Suction-pipe connections 2 to 4 inches greater diameter.

Cushion-valves better arranged.

Air-chamber is made much larger.

Shells and bolting are warranted especially strong.

The following necessary fittings, heretofore charged for as extras, are included in the price, and regularly furnished as a part of this pump,—viz.:

A capacity-plate is added.

A stroke-gauge is added.

A vacuum-chamber is added.

Two best quality pressure-gauges.

A water-relief valve of large capacity.

A set of brass priming-pipes and valves.

From two to four Chapman hose-valves.

A sight-feed cylinder-lubricator is added.

Respectfully submitted,

JOHN R. FREEMAN,

Engineer Assoc. Mutual Ins. Co.

June 1, 1891.

The following vote was passed at the monthly conference of the Associated Companies, June 15, 1891:

"Voted: To recommend that the so-called 'Underwriter's Pattern' of steam fire-pump, complete with all attachments as per specification of June 1, be recognized as the approved type, and that pumps built and fitted up less perfectly than per this specification be not approved in future installations, except under special circumstances and by special agreement with some executive officer of the insurance companies."

CHAPTER XIV.

MINING-PUMPS.

Pumping engines for mines are generally placed underground; there are instances in this country in which the engines are located on the surface, but by far the larger number are down in the mines, and usually making the delivery in a single lift. The service required of a mine-pump is constant, and as pumps are not always duplicated, it is a matter of first importance that they be properly proportioned as to strength, and furnished with a valve mechanism not liable to get out of order.

There was formerly no uniformity in the type of pump used in mines; piston-pumps and several varieties of packed plunger-pumps were employed, but, all things considered, the use of plunger-pumps is preferable to that of piston-pumps, especially in deep mining operations. The plunger-pump is almost exclusively used at this time, for in the old plan of working under a heavy pressure with a piston, if dirty water were met with, the pump is less likely to give satisfaction; the cylinder cuts out quickly, and if a lining is employed, it is a costly operation in a large pump to put in another.

The metal used for mine-pumps is almost always hard cast iron; occasionally, however, a gun-metal water-end is used, but such examples are rare, especially for large pumps. It is customary to give water-end castings for mine-pumps an extra thickness of metal apart from that necessary for working strength, including any jar or water hammer, the pump castings under pressure having from half an inch to an inch and a half added to the thickness for deterioration. The metal

itself is made very hard; in fact, as hard as can be worked with special steel tools; ordinary machine castings have been found to be entirely too soft to long withstand the action of bad water. A water-end of a pump was recently shown the writer, in which, by a sudden change in the character of the water in a mine from sweet to bad, the pump was utterly ruined in less than a month's service. The pump was brass fitted, the castings soft gray iron; the action of the acid water was to loosen the valve-seats and attack every interior portion of the pump which came in contact with the brass, as well as corroding such portions as had been machine fitted.

The action of mine water is very capricious, and does not affect all portions of a pump alike; that is to say, all parts are subject to the destructive action of the acid water, but certain localized points in the interior are often found to deteriorate much more rapidly than others. For such portions of a pump special provision may be made in advance by having duplicate parts, if the water-end be made up of sections, easily attached whenever needed.

Sinking-pumps are usually vertical direct-acting single pumps with steam-thrown valve-gear. The water-ends are of either the piston, differential plunger, or fitted with double-acting plungers centrally packed; the latter, being a better type of pump for handling gritty water, is more favorably thought of by mine-operators than either of the former. Such pumps are fitted with an eye-bolt in the upper cylinder-head, or with three eye-bolts attached at points of support, which will enable the pump to hang vertically when suspended by a chain. In addition to this, wrought-iron dogs or clamps are provided by which the pump may be fastened to suitable timbers on the side of a shaft; the pump may, therefore, be either held in suspension from above, and raised or lowered as may be required in the case of a flooded mine, or it may be temporarily fixed to the side of the shaft, to be lowered at such intervals as the progress of sinking a shaft may require.

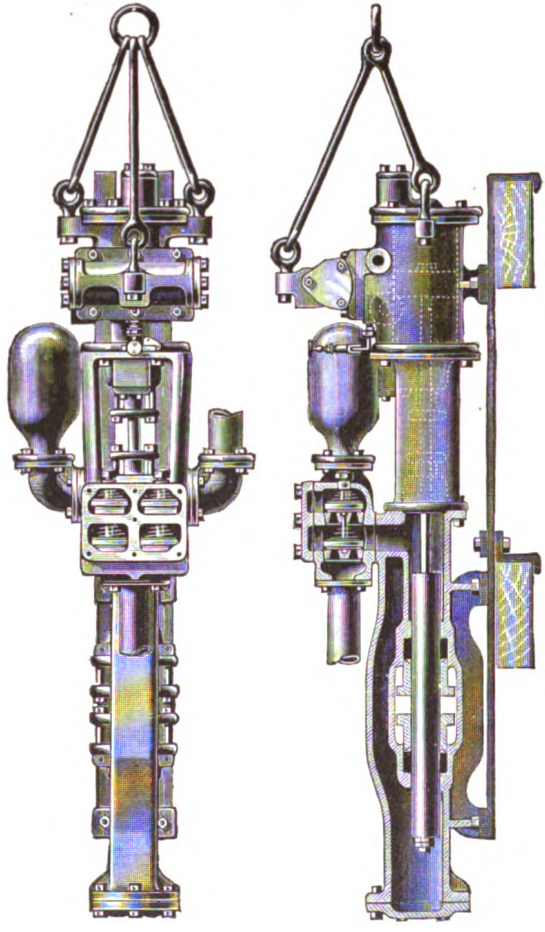
A sinking-pump must be simple, compact, strongly made, and the valve-gear well protected. Probably no other type of pump is subjected to the rough usage which sinking-pumps invariably get. It is a condition which cannot be overcome in the preliminary stages of mining, on account of the liability of accident from blasting operations, the necessity at all times for pumping gritty and often acid water, and the fact that there is always more or less muddy water trickling down the shaft from the water-bearing strata overhead, which so completely covers a pump with mud that if it were not in motion it could scarcely be distinguished from any other mass of débris. Notwithstanding all this, a sinking-pump must work continuously night and day, and often up to its limit of capacity. The failure of such a pump at a critical period, even for a day, may flood a shaft, which would require a week or more to recover.

The Cameron sinking-pump, by reason of its having no external valve-gear, has met the above conditions, and as a result it is very popular with mine-operators. The steam-end of this pump has been previously described and illustrated in the sectional engraving, Fig. 160, which shows the mechanism by which the steam-piston is operated for horizontal mine-pumps ; for vertical sinking-pumps the valve-chest is placed at right angles to the stroke, the port-openings being arranged to correspond, a detail easily understood, and does not require special illustration. A sectional elevation of a double-plunger pump, showing the arrangement of stuffing boxes, suction- and delivery-valves, and suction-pipe are clearly shown in Fig. 194.

A vertical sinking-pump by the Deane Steam-Pump Company is shown in sectional elevation in Figs. 195 and 196. This design is intended to meet the demands for such a machine in the mining districts of Colorado. These machines are for sinking or unwatering mines or for any use where a varying water-level must be accommodated. The plungers are in one piece, are double-acting and centre-packed; they can be very easily

and quickly packed by the removal of only two glands. The valves are of rubber, of a special texture for heavy duty and to resist the action of bad mine water, with brass covers.

FIG. 194.



Hinged bolts are used on the hand-hole plates; stuffing-box glands and all removable parts and all nuts throughout the machine are secured by split pins, so that no parts may be lost while the machine is being packed or adjusted. The

machine is fully equipped with drip- and sand-valves. The suction is taken in at the bottom of the pump, thus saving a length of pipe required by most machines of this class. The

FIG. 195.

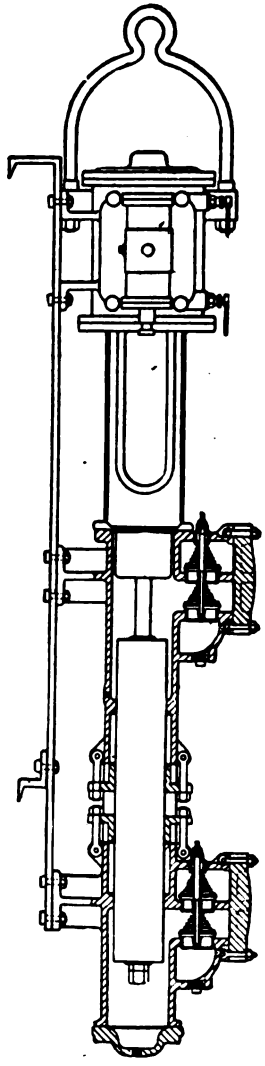
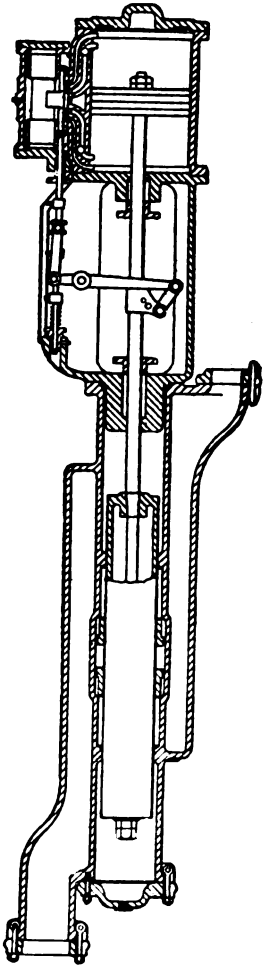


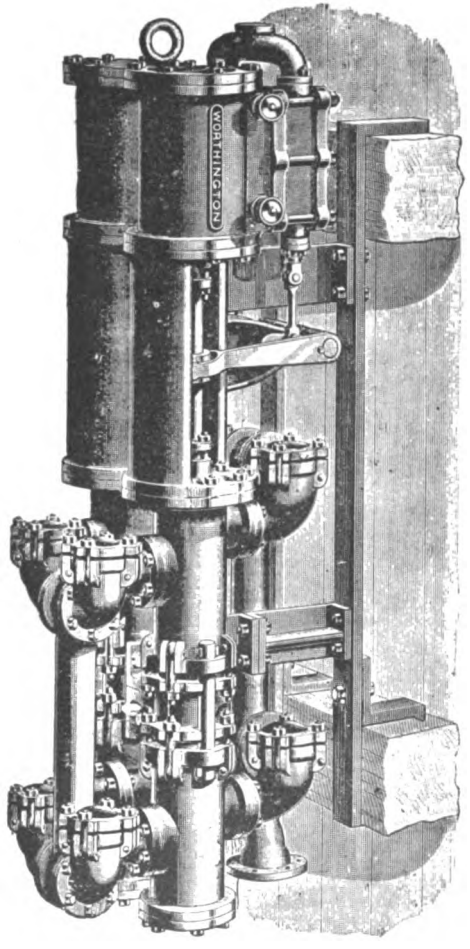
FIG. 196.



valve motion is the well-known Deane patent. It is absolutely positive, and will start at any point of its stroke, and is thor-

oughly protected by a cast-iron shield, which also serves as the yoke between the steam- and water-ends. The steam-cylinder is equipped with cushion-valves to regulate the up and down strokes of the engine. Each machine is made with a bale attached to the steam-cylinder, and also with shoes and dogs to secure easy handling and control while in the shaft.

FIG. 197.



A duplex sinking-pump by Worthington is shown in Fig. 197. It is not often that duplex pumps are arranged to work vertically, but when properly designed for such a position they will be quite as satisfactory in their operation as if employed horizontally. A close inspection of the engraving of this sinking-pump shows that in the several details of the water-end the best practice in mining-pump design has been carried out. The water-plungers are double-acting, working through exterior stuffing-boxes and adjustable packing. Means are provided, as shown in the engraving,

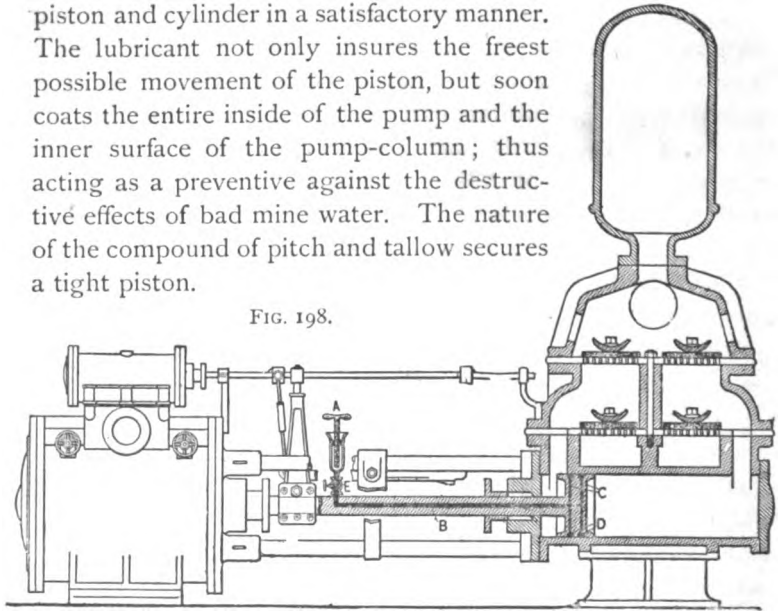
for either suspending the machine at the eye-bolt shown between the steam-cylinders, or for hanging it on suitable timbers on the sides of the shaft. The suction opening is at the lower end of the pump, convenient for attaching the suction-pipe or hose. The discharge connection to column-pipe is on the side. The water-valves are enclosed in heavy pot-chambers, and are made accessible for examination or repairs by means of swing-bolt covers on the valve-pots. The duplex valve-movement being positive, the pump is always ready to start, and when running there is an entire absence of concussive action; it is, therefore, well fitted to withstand the rough usage to which sinking-pumps are subjected.

Single direct-acting mine-pumps occupy less room in a mine than a duplex pump of the same capacity, but the delivery from a single pump is at times so faulty that a very large percentage of the steam used is expended in starting anew the column of water at each stroke. This fault is due, in part, to the construction of the pump, by which an interval occurs at the end of each stroke, during which no water is being delivered, the flow is suspended for the time, and thus permits the water column to come to a state of rest. But the fault does not lie wholly with the pump as such, but to neglect in the failure to keep the air-chamber of the water-end properly charged with air. If this latter detail were properly looked after there would be less trouble with pumps and valves than has heretofore been the case.

The capacity of an air-chamber in a single, direct-acting mine-pump should never be less than four volumes of the pump capacity per stroke. This air-chamber should connect directly with the highest point of the delivery-chamber; it should be furnished with a glass water-gauge so that the quantity of air in the vessel can be determined at a glance. The height of the water in the air-chamber should never exceed a level due to one-third of its cubic content. A device similar to that shown in Fig. 91 can be applied to any pump, and the deeper the mine the greater the necessity for its application.

Lubricating Pistons of Mining-Pumps.—The sectional elevation, Fig. 198, represents one of the Knowles piston mining-pumps, fitted with Simpson's patent lubricating water-piston. This device consists of an arrangement whereby a lubricant composed of pitch and tallow is sent from the inside of the pump-piston through its rings to the surface of the pump-cylinder, thus lubricating the water-piston and cylinder in a satisfactory manner. The lubricant not only insures the freest possible movement of the piston, but soon coats the entire inside of the pump and the inner surface of the pump-column; thus acting as a preventive against the destructive effects of bad mine water. The nature of the compound of pitch and tallow secures a tight piston.

FIG. 198.



The action of this device is as follows: That part of the piston-rod which enters the water-end of the pump has through its centre a long passage-way, *B*, which connects the small hand-pump *A* holding the lubricant with the holes *C* in the pump-piston, as shown in the engraving. The holes *C* are drilled in the water-piston at several points, ending under the packing-rings *D*. The packing-rings are made of composition copper and tin, and are perforated with small holes to permit the compound of pitch and tallow to ooze out and lubricate the cylinder and piston. The lubricant is by this little hand-pump forced through the hollow piston-rod and the holes in the

water-piston, thus filling them and setting up the rings in the water-piston. When the lubricant is pumped up solid, cock *E* is closed to retain the pressure. In the action of the pump a small amount of pitch and tallow oozes out through the holes of the packing-rings, lubricating the cylinder and producing the benefits above named. It is only necessary to charge the pump with this lubricant about every eight hours.

Duplex mine-pumps are steadily growing in favor, because the duplex valve-movement is particularly well suited to handling water at high pressures. The advantages of this valve-movement have already been set forth in the chapter on hydraulic-pressure pumps, and need not, therefore, be repeated here.

Pumps of this type are always placed at or near the bottom of the mine, and no difficulty is had in making the delivery in a single lift, wholly irrespective of the depth of the mine.

Air-chambers are not usually supplied duplex mine-pumps; effective and quiet working having always accompanied the practical operation of pumps of this type, it was believed that the addition of an air-vessel was unnecessary. This is true in part, but the duplex action is not wholly continuous, or perhaps a better way to put it is that the pressure in the delivery-pipe is not absolutely uniform, but it is more nearly so than in any other type of pump now employed in mining operations. The addition of an air-chamber, together with an automatic charging device, as shown in Fig. 102, will fully meet all the conditions of this exacting service.

The superiority of the duplex movement was strikingly exhibited in the development of the pipe-line system of the National Transit Company, the service being analogous to mine-pumping except as to material. When single direct-acting engines were used the jarring effect produced by their irregular delivery was very destructive on the pipes and fittings, causing leakage, and was altogether a constant source of expense in the maintenance of the pipes, to say nothing of the losses occasioned by the delay in making oil deliveries.

The substitution of duplex for single pumping engines corrected at once this troublesome detail in the pipe-line business. A 12″ × 48″ duplex pumping engine was built by Worthington for the National Transit Company some six years ago, which has ever since its erection been at work, continuously delivering 25,000 barrels of oil (50 gallons) in 24 hours at a pressure of 1500 pounds per square inch, pumping the oil over a mountain of a height corresponding to the above pressure.

So general is the acknowledged superiority of this type of pumping engine for mining operations over the single direct-acting pump, that many firms whose reputation and success were wholly built up in the single pump business are now supplying their patrons with pumps having duplex steam-ends, either simple or compound.

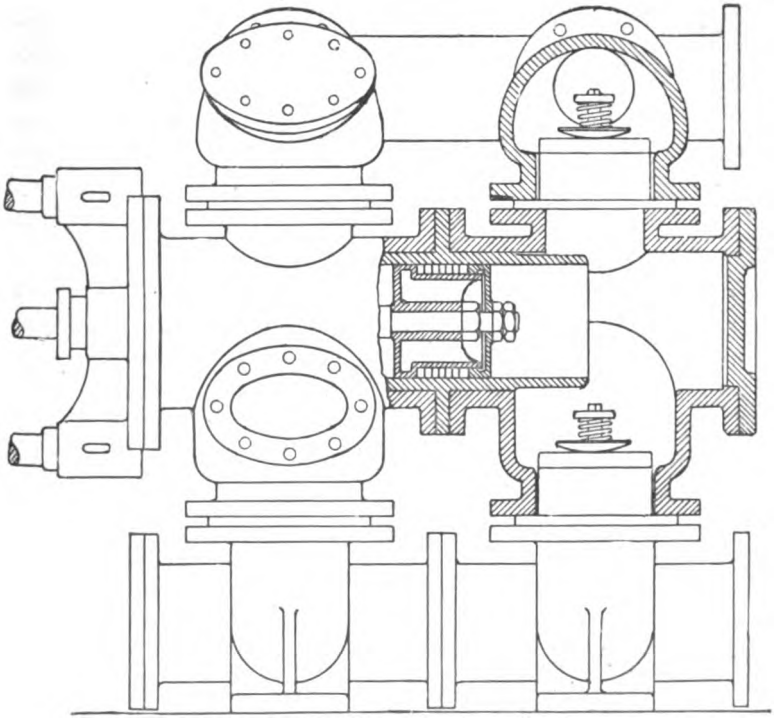
A piston-pump with gun-metal lining, such as shown in Fig. 113, is often used in mines where the depth does not exceed two hundred feet and the water is not acid. The fibrous packing can be readily renewed whenever necessary. Removable gun-metal linings are commonly arranged so that they can be partially turned in the water-cylinder to present a new or uneven surface underneath the piston. As the bolt-holes are spaced on either three or four divisions of the circumference, a corresponding new surface may be brought under the piston until it is so completely worn as to need renewal, in which case the substitution of a new lining for the one worn out can be accomplished in a few minutes.

Piston-pumps occupy less space in a mine than plunger-pumps of the same capacity. If the water is not bad, the pump may be of the same general construction as for any other pumping service for which a piston-pump would be selected. It is a wise precaution, however, to have the pump brass-fitted throughout; it adds but little to the cost and is well worth the difference.

Piston-pumps for acid water should have the water-ends made up of sections, any one of which may be easily

detached from the assembled water-end and replaced by a new one, in case such renewal should become necessary through the corrosive action of the mine water. Fig. 199 represents partly in section and partly in elevation one side of a $7'' \times 12''$ duplex water-end, from designs by the writer. The suction-pipe underneath is made in two pieces, with a flanged joint in the centre; these two castings are in all respects alike; they ex-

FIG. 199.



tend across the pump similar to those in Fig. 201. There are four water-cylinders precisely alike. The working barrel of the pump is made of gun-metal, and fits into an opening, as shown in the drawing; the flange included in the gun-metal barrel is faced and drilled similarly to those of the two water-cylinders to which it is applied, through-going bolts securing

all together. The valve-seat is shown in elevation in the above drawing; a section of a similar valve-seat is shown in Fig. 200. The width of the bottom flange is only sufficient to make a good joint. It will be seen that the water-cylinders, suction- and delivery-pipes are simply faced castings, held together by through-going bolts.

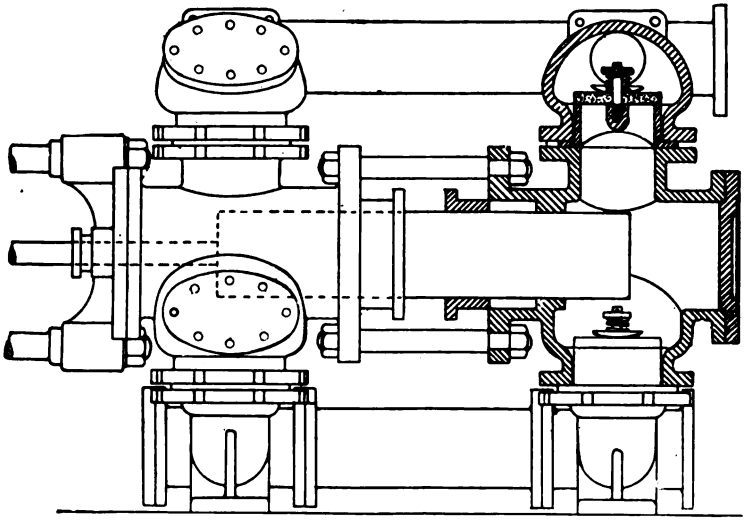
Centrally-packed plunger-pumps, similar in general design to the 8″ × 12″ duplex shown in Fig. 200, are largely used in anthracite coal-mines. The water-cylinders are constructed with special reference to pumping acid water. The shape is made up of curved surfaces as far as practicable. There is no fitting to these chambers other than the facing and drilling of the flanges, and the boring of the stuffing-box; the interior surface or skin of the casting not being disturbed offers a better resistance against the action of bad water than such parts as have had this surface removed.

The plungers are made of cast iron, working through a stuffing-box included in the water-cylinder casting. These stuffing-boxes and glands are sometimes lined with gun-metal bushings, as shown in Fig. 37, but not as a general practice. The plunger is always kept well lubricated, which greatly retards the corrosive action of the water upon it. This lubricant also covers the interior surfaces of the gland and the bored guide at the bottom of the stuffing-box, so that corrosion proceeds less rapidly there than elsewhere in the cylinder. In this design the plunger is driven by a rod passing through a stuffing-box in the inside head, as must be the case in centrally-packed pumps.

This rod should be of some copper and tin composition, and never of Muntz-metal. The writer has used Tobin and phosphor-bronze with satisfactory results. This gland and stuffing-box should be fitted with gun-metal bushings as shown in Fig. 37.

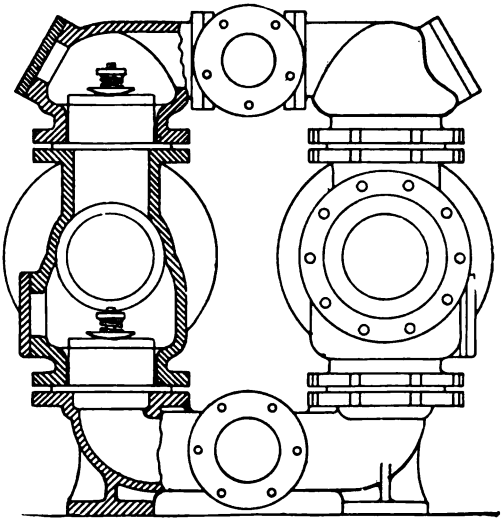
The valves are usually one single rubber disk for plungers up to 12 inches diameter, and it is this diameter which usually fixes that of the valve, both being alike. The valve-seats being

FIG. 200.



made of hard gun-metal will outlast several water-ends; the flange by which a seat is held in place need not be wider than will make a safe joint. The delivery-valve chamber is of a form well calculated to resist the action of the current of water passing through it to the central delivery-pipe shown in the end elevation, Fig. 201.

FIG. 201.



A duplex centrally-packed water-end for a 15″ × 36″ mine-pump, from designs by the author, are shown in Figs. 202 and 203. It does not differ

Digitized by Google

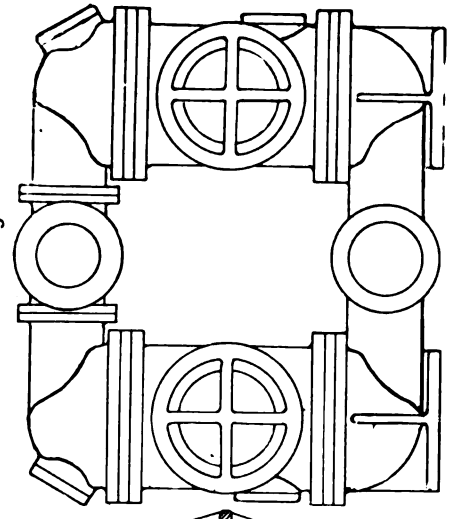


FIG. 203.

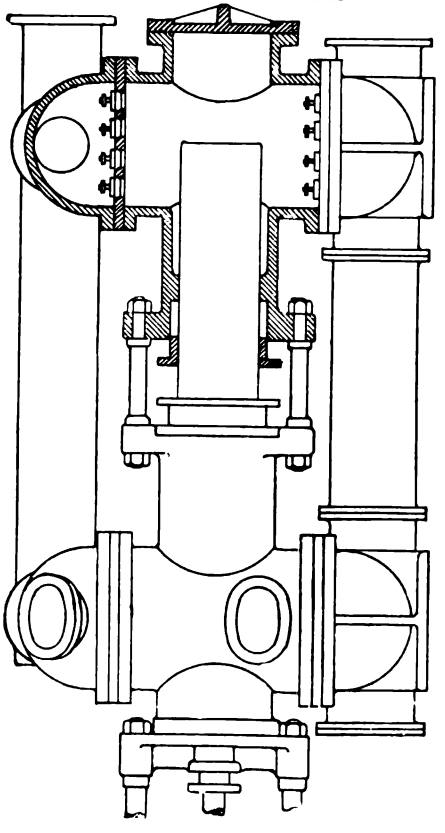


FIG. 202.

materially from the pump just described except in the fact of its having a gun-metal valve-plate fitted with as many small valves of the design shown in Fig. 55 as the plate will contain. The diameter of this plate can usually be made to accommodate from 60 to 75 per cent. of plunger area, through valve-seats 3½ to 4 inches in diameter, without enlarging the water-chamber to an unreasonable proportion. By reason of the uncertain demands upon mine-pumps it is recommended that for strokes of 18 to 36 inches the combined area of valve openings be not less than 60 per cent. for the shorter stroke, and from 75 to 100 per cent. for the long stroke, as high plunger speed should always be

accompanied by a correspondingly large valve area to get the best results.

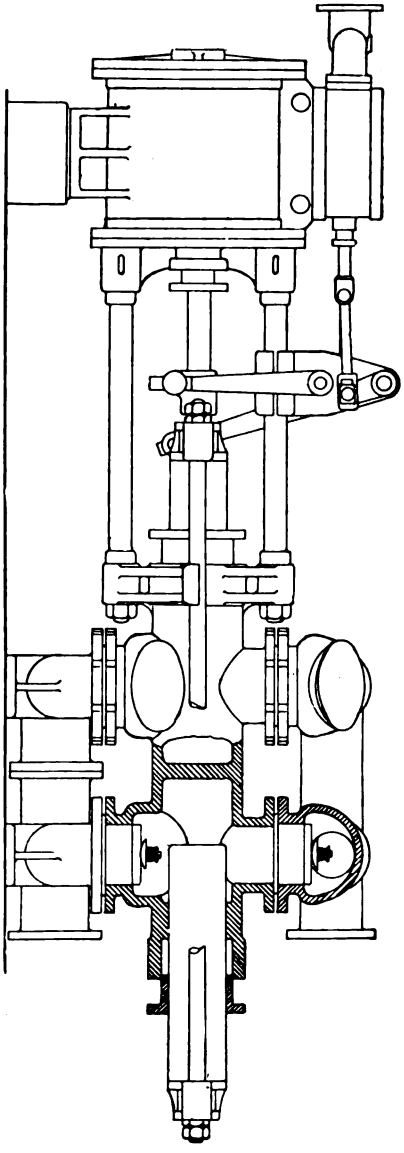


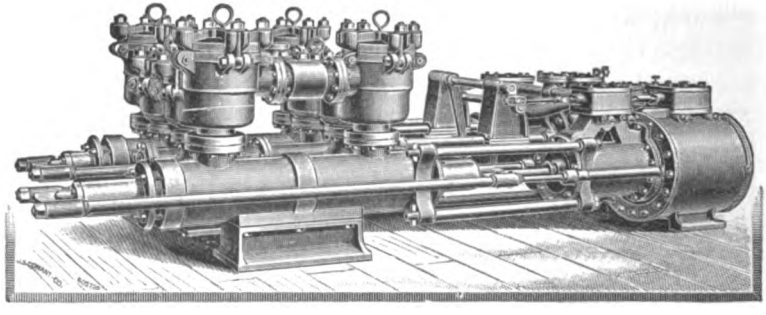
FIG 204.

An outside-packed plunger-pump with end stuffing-boxes and central diaphragm, as shown in Fig. 204, is an excellent form of mine-pump. The absence of a piston-rod working through one of the water-cylinder ends is an advantage. The illustration represents one side of a 9″ × 18″ water-end; by reason of the projecting plungers this pump requires a greater length of floor space for the same working stroke than those having centrally-packed plungers. The end stuffing-boxes are more convenient for filling and for adjustment than when placed centrally. Each plunger has a cross-head included in the same casting; wrought-iron or steel tie-rods connect each pair of plungers belonging to one side, as shown in the drawing. The valves, valve-seats, valve-chambers, suction-

and delivery-pipes are substantially the same as for the two preceding pumps. Each water-cylinder casting includes two plunger-chambers; the lugs for attaching the steam-end being included also, it will be seen that there is much less fitting required in this than is called for in centrally-packed water-ends; there is one objection, however, which must not be overlooked, and that is, in the event of any portion of a water-cylinder of this design being sufficiently affected by bad water as to require renewal, a complete water-cylinder must be furnished.

A compound mine-pump by the Buffalo Steam-Pump Company is shown in perspective view in Fig. 205, a side eleva-

FIG. 205.



tion in Fig. 206, and an end elevation in Fig. 207. The latter two illustrations are sufficiently sectioned to show the arrangement of plunger, pot-valve chamber, suction- and delivery-valves.

The high-pressure steam-cylinders of the pump here shown are 12 inches diameter; low-pressure cylinders, 22 inches diameter; stroke, 18 inches; the plungers are 7 inches diameter. The rated capacity is 300 to 350 gallons per minute, delivered 600 feet above the station with a steam-pressure of 65 pounds in the initial steam-chest. The plungers are tied together with steel tie-rods coupled into cast-steel cross-heads. The plungers are carried through bronze-lined stuffing-box glands. The water passages are 50 per cent. of the plunger

area. Each valve-chamber contains three valves of a size that also equals 50 per cent. of the plunger area; both suction- and discharge-valves are overhead, so that the plungers are always water-packed. The caps of the valve-chambers are held by swing-bolts. The pump is supplied through a 7-inch suction-pipe, and the discharge is driven through a 6-inch column. This pump was furnished with an improved single air-pump and condenser designed for the most severe duty. The valves of the steam-cylinder are of the ordinary plain slide type, and arranged to run by the ordinary duplex movement already described.

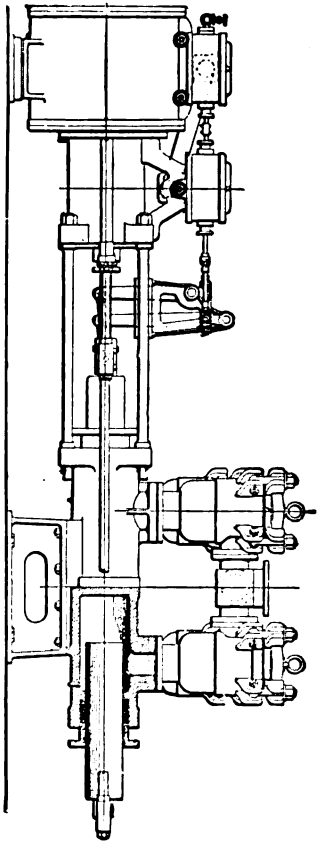


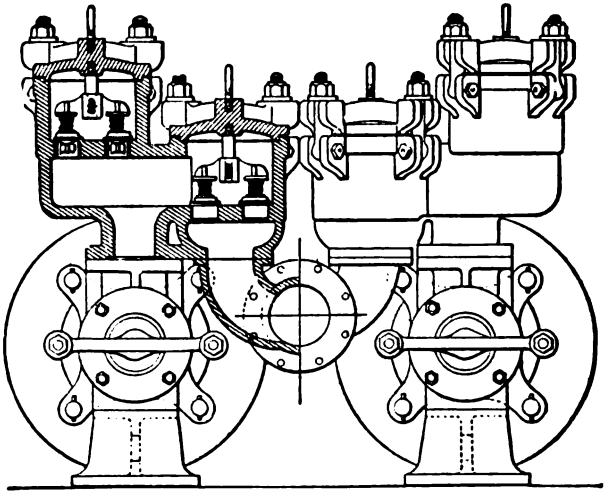
FIG. 206.

A Knowles compound condensing duplex plunger-pump, illustrated in Fig. 208, is from a drawing representing the 800-foot station of a mine, with the pumping engine in place and the sinking-pump in position below the station. This pump has a capacity of 1000 gallons of water per minute. The water-end is fitted with pot-valve chambers similar to those described in the chapter on hydraulic-pressure pumps. This pump is arranged to draw its supply from the sump located underneath the engine, one or more sinking-pumps raising the water from a lower level. From this same tank the condensing apparatus draws its supply of injection water necessary for condensing the exhaust steam as it issues from the low-pressure cylinder. The water from the hot-well delivery of the

condensing apparatus flows back into the sump from which it was taken, an arrangement of piping making this delivery near to the main pump-suction, so that the hot-well discharge is at once taken up by the main suction instead of communicating its heat to the whole body of water; by this means a large volume of water is available for condensation which has received no heat from the hot-well discharge.

The engraving shows a float in the sump; this float operates a balanced throttle-valve on the steam supply-pipe. The movement of this float, up or down, regulates the speed of

FIG. 207.

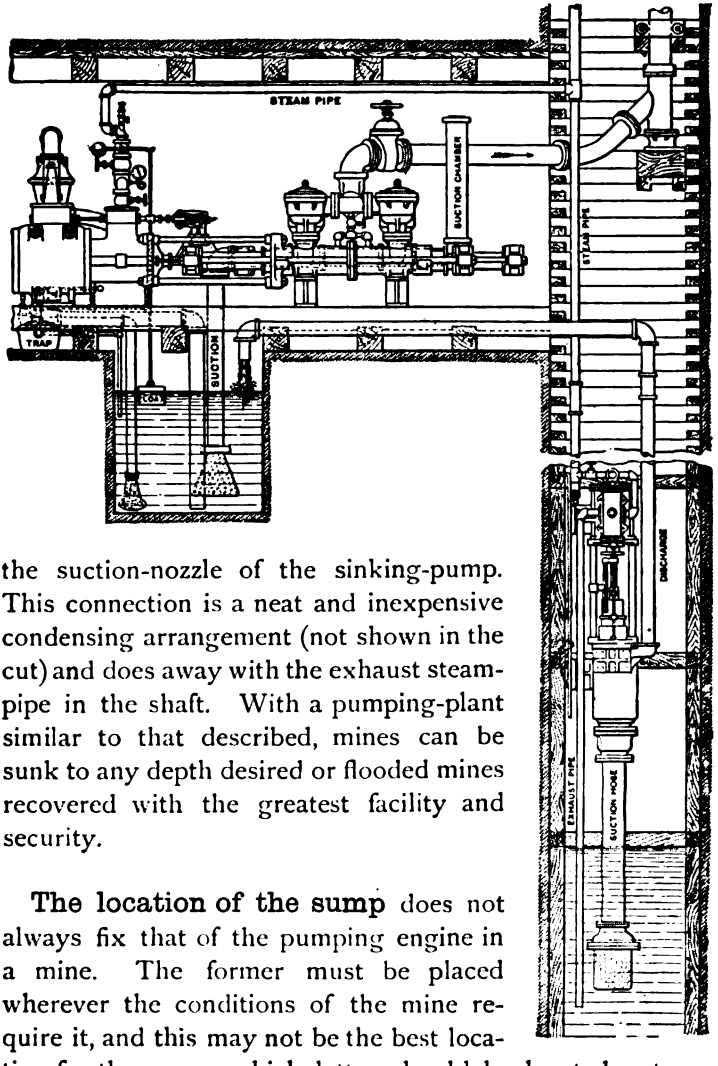


the pumping engine, and will stop the engine automatically should the sump be emptied of water.

In deepening the shaft the sinking-pump is lowered at intervals as the excavating progresses. The water at the bottom of the shaft is taken by the sinking-pump and delivered to the tank or sump that supplies the pumping engine in the station. A single steam-pipe down the shaft supplies both the pumping engine and sinking-pump. The exhaust steam from the pumping engine, as before explained, is condensed by the independent air-pump and jet-condenser. The

exhaust steam from the sinking-pump, however, is usually carried to the sump or into a special connection arranged on

FIG. 208.



the suction-nozzle of the sinking-pump. This connection is a neat and inexpensive condensing arrangement (not shown in the cut) and does away with the exhaust steam-pipe in the shaft. With a pumping-plant similar to that described, mines can be sunk to any depth desired or flooded mines recovered with the greatest facility and security.

The location of the sump does not always fix that of the pumping engine in a mine. The former must be placed wherever the conditions of the mine require it, and this may not be the best location for the pump, which latter should be located not more than 20 feet above the bottom of the sump; and as for the

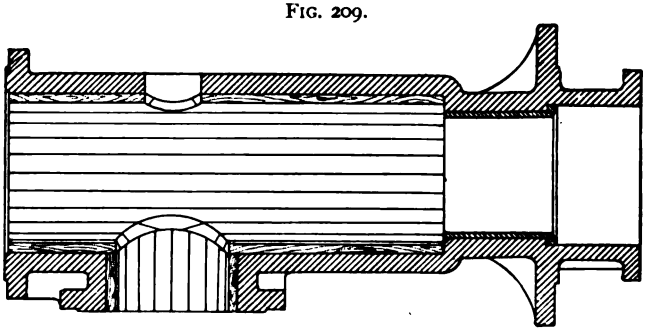
length of the suction-pipe, it is well understood that it should be as short as possible. This is a detail in mine engineering which we cannot enter into here, but the main pumping engine, if more than one is required, ought to be placed as near the shaft as possible, and if water has to be pumped from several distant points, and perhaps at greatly varying levels, small, low-service pumps may advantageously be used at any convenient place in the mine, the combined deliveries of these pumps being into a sump at a convenient level for the main engine; an arrangement such as that shown in Fig. 208, reproduced from Knowles's catalogue, will make our meaning clear.

Lining a Mine-Pump with Gun-Metal.—The experiment has been tried of lining pump-barrels and other interior portions of mine-pumps with gun-metal, but it has proved a costly thing to do. The intricate details of a pump interior prevents lining the entire surface with gun-metal. The only practical method by which it can be done is to have all portions cylindrical, so that the fitting would be reduced to the simple operations of boring and turning, and then forcing them in place. By the time all this is accomplished the cost will fully reach that of a properly-designed water-end constructed wholly of gun-metal.

Lining a mine-pump with wood is better and cheaper than with gun-metal. The pump details must have been previously arranged with reference to the insertion of the wood lining. Fig. 209 represents a water-cylinder of a horizontal plunger-pump with the wood lining in place. The staves are of soft pine, machine-dressed radially and to the outer and inner curves, suited to their respective diameters; they are cut to exact length and arranged in place. Two of the staves are then bevelled to admit a third stave between them, also bevelled, and which shall act as a wedge; this middle stave is then driven home with a maul; all the staves will now be firmly fixed in their places in the interior of the cylinder. Any

openings into this cylinder may be lined in the same manner, care being taken that the inner ends of such staves closely fit the curvature of the main staves; after these are securely wedged in place the opening may then be cut through into the working-barrel. A gun-metal bushing is shown inserted

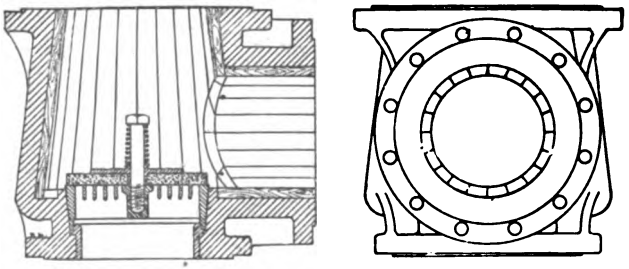
FIG. 209.



in the bottom of the stuffing-box, through which a cast-iron plunger works.

A valve-chamber, shown in Fig. 210, is similarly lined with wood. It will be observed that this interior is an inverted frustrum or a cone with a cylindrical intersection. The inser-

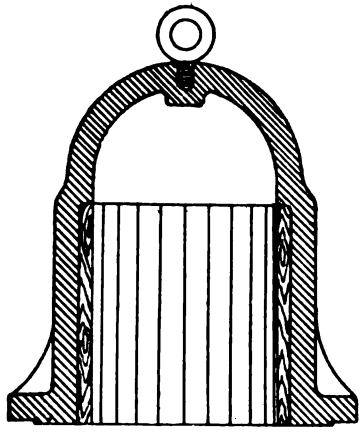
FIG. 210.



tion of this lining is not so simple a matter as that of a plain cylinder, because each stave requires to be fitted to the two top and bottom diameters of the valve-chambers. By making a wooden templet, having the proper curves and angles accurately representing one stave, duplicates can be quickly made

by a wood-worker, each piece exactly fitting into its place. Three staves must be fitted for wedging, as in the preceding paragraph. After the conical portion has been lined, the cylindrical intersection can then be fitted, after which the branch opening may be cut through. This chamber is fitted with a gun-metal lining below the valve-seat. The valve-seat is inserted on a taper, and extends down, forming a lining to protect the metal at the top of the bushing. It will be seen that this chamber is very thoroughly protected.

FIG. 211.



An air-chamber with wood lining is shown in sectional elevation in Fig. 211. Only that portion is protected that comes in contact with the water.

The duplex water-end, Fig. 212, is lined throughout with wood, as shown in the preceding illustrations, and in end elevation in Fig. 213. This pump is fitted with plungers 12 inches diameter by 48 inches stroke; it is located in an anthracite coal-mine nearly 1000 feet deep. The whole design is good, and may be said to fairly represent the best modern design and construction, except in two points, about which engineers are not fully agreed: one, in regard to a centrally-packed plunger, as against end-plungers; and the other, in the employment of one large valve instead of several smaller ones. The thickness of metal in this water-end subject to pressure is two inches; the wood lining one inch thick. The valves are 15 inches diameter and 1¼ inches thick. No air-chambers are provided except the ones over each delivery valve-chamber.

The Cornish pumping engine was among the first pumps used in the anthracite coal regions of Pennsylvania. The late Howell Green, whose experience with mining-pumps

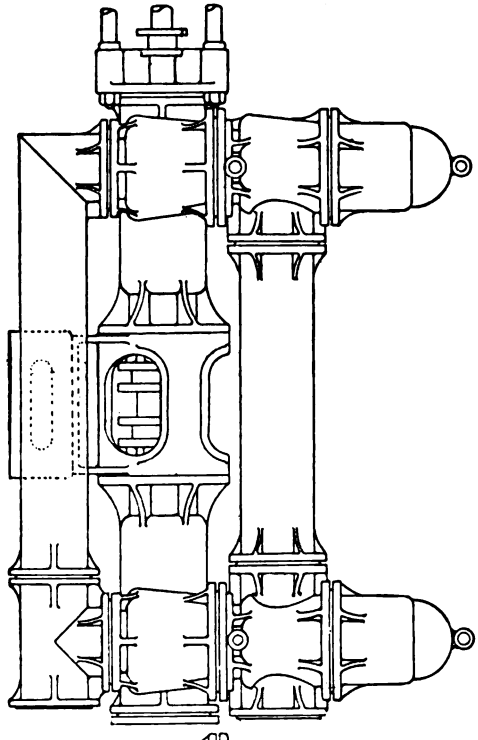


FIG. 212.

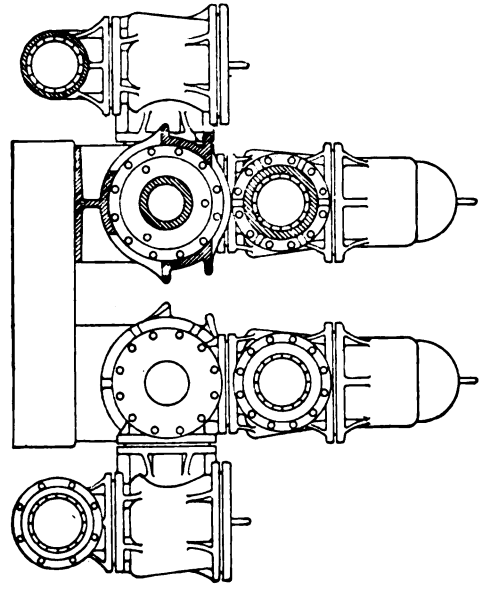


FIG. 213.

was both varied and extensive, was of the opinion that "if the working-barrel was made of hard iron and the length of rod nicely adjusted so that the leather would come slightly above the bore in the upstroke, and if the water was clean and not acid, and the lift not too heavy—then, it is still the best mining-pump ever made." This view is not now entertained by mining engineers, and the types of pumping engines already described and illustrated in this chapter are, for many reasons, to be preferred.

That type of pumping engine known as the Cornish may be described as a single-acting, high-pressure, expansive condensing engine, working single-acting pumps through the medium of a beam. Cornish pumps are usually of the plunger pattern, the plungers being loaded with iron weights sufficient to counterpoise the pressure of the water column. The engine may be considered as consisting of two parts; the power of the engine is used to lift the loaded plunger, after which the steam-engine part of the machine is detached and the weighted plunger is allowed to descend by gravity at a speed depending on the quantity of engine-power in action and the rate at which the water is being drawn away. The chamber of the pump becomes filled when the plunger is raised. and the act of inhaling the full charge through the suction-valve is a portion of the work which the steam has to perform, and a portion also much subject to variation.

The speed of the engine is regulated by an adjustable cataract; the exhaust-valve first and then the steam-valve are thrown open by treadle-weights, as soon as the catches are detached by the cataract. The valves are closed by tappets on a plug-rod, first the steam-valve and then the exhaust-valve, the former at a period of the stroke varying in practice between one-third and one-fifth from the commencement, and the latter at the end of the stroke.

In engines working on this principle, as also in all reciprocating engines pumping without cranks, there is nothing to limit the strokes of the engine to any exact length. It is necessary, therefore, that bumpers or catch-pieces be provided

to restrain the engine at both ends from an undue length of stroke; and thick plates of india-rubber under hard-wood blocks are now used for this purpose in place of the spring beams formerly employed. An engine thus arranged, working alone, lifting water from one fixed level to another, would work continuously with one length of stroke and at one speed, at whatever it might be set.

The single-acting engine on the Cornish principle was thought to have some advantages over a pumping engine with crank and fly-wheel, in the fact that no power is required in the Cornish engine for keeping gearing in motion at each end of the stroke; a certain amount of percussion action is indeed necessary to overcome the inertia of the engine at the beginning of the stroke; but, on the other hand, the whole engine is brought to a dead stand at the end of every stroke by the whole effective power being completely absorbed in the work done in pumping.

The Jeanesville Mine-Pump.—The compound duplex mine-pump illustrated in Fig. 214 is from designs by Mr. Vernon H. Rood, and constructed by the Jeanesville Iron-Works, Jeanesville, Pa. It represents their standard design of compound condensing, duplex, outside-packed plunger mine-pump, designed especially for the anthracite and bituminous coal-mines of Pennsylvania.

The pump selected for illustration is one having—

High-pressure cylinders, 25 inches diameter.

Low-pressure cylinders, 42 inches diameter.

Pump-plungers, 14 inches diameter.

All of 48 inches stroke.

Its rated capacity is 2000 gallons of water per minute against a head of 425 feet.

The steam-end shown in sectional elevation in Fig. 215 is of the ordinary duplex form. One rod is common to both pistons, and passes through stuffing-boxes in the high- and low-pressure cylinder-heads. The valve-faces of both cylinders are in the same plane, the valves being operated by a

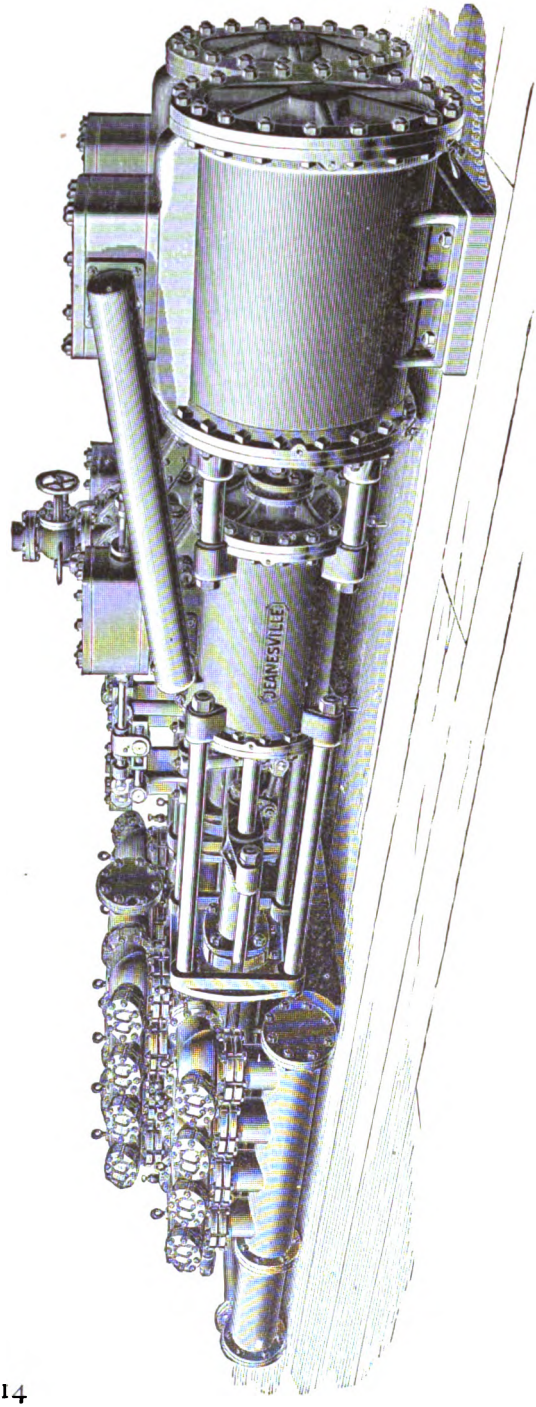


FIG. 214.

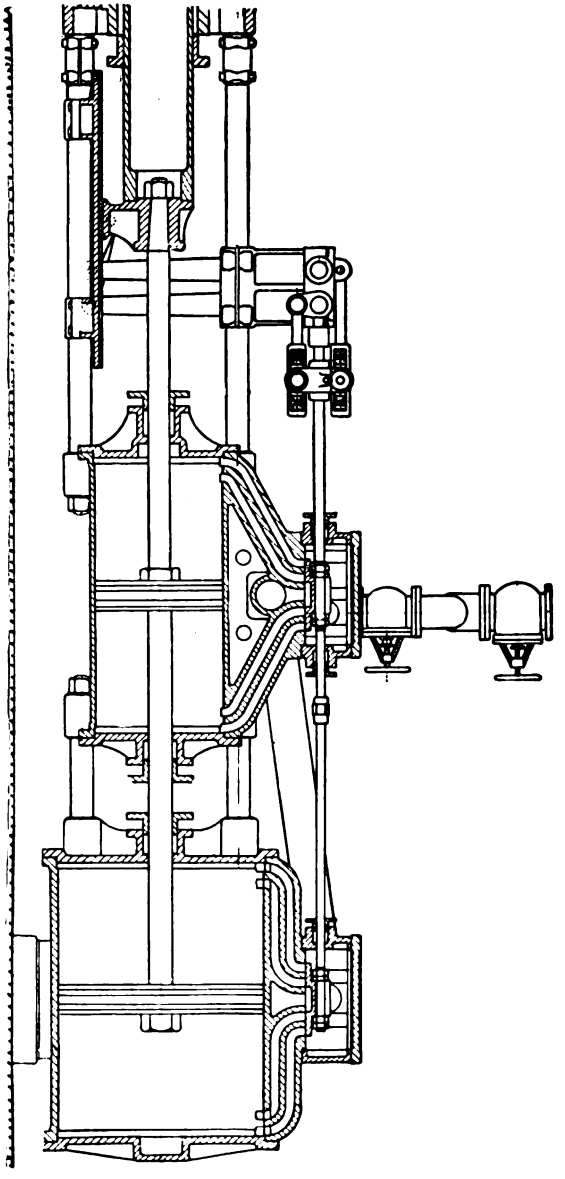
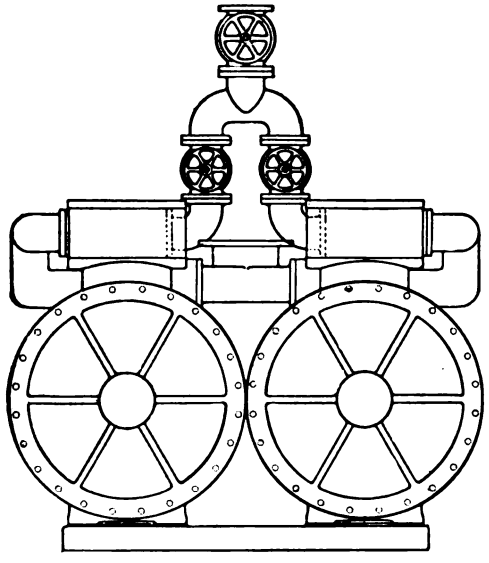


FIG 215.

driving mechanism similar to that shown in Fig. 178. The dash-relief valves are fitted to the high-pressure cylinder only. This detail is similar to that of Fig. 173. An end elevation of the steam-cylinders is shown in Fig. 216. The customary practice on all large compound duplex steam-ends in having one main and two auxiliary throttle-valves is here shown. The object of the auxiliary throttle-valves is to give each engine a separate adjustment should one side appear to work a little slower than the other; in starting and stopping the engines the upper or main valve only is used, the auxiliary valves not being disturbed in their adjustment. This steam-end, in common with the general practice in mining-pumps, is not steam-jacketed, but when in position in the mine and thoroughly tested, the cylinders and chests are covered with a non-conducting material, such as magnesia, which can be laid on with a trowel. This prevents radiation and a protection against the water in a "drowned-out" mine. The steam-ends of mine-pumps are generally made of unusual strength, with extra large wearing surfaces and connections to withstand the continuous and severe duty they are called upon to perform in times of high water, or possibly in a "drowned-out" mine, when it will be forced to work submerged to a depth of from 10 to 60 feet, in which case lubrication or any other care is impossible.

FIG. 216.



The water-end is shown in partial sectional elevation in Fig. 217. It consists of four cylinders, or working-barrels, as they are generally called, bolted together in pairs, with a recessed blank flange between them; these working-barrels rest on two cross-feet, which serve to tie the whole four together as if they were one casting. To reduce the number of spare working-barrels necessary to keep on hand, and also to facilitate their removal when worn out, the bosses for receiving the tie-rods which connect the pump to the steam-cylinders are not cast to the water-cylinder, but to a strongly-bracketed flange, which is bored and faced to fit over the outside of the stuffing-box and against a flange cast on the working-barrel from 8 inches to 10 inches from its end, as shown in the drawing.

These pumps, in accordance with the best modern practice, are built to gauges, so that only two spare working-barrels are required for a whole pump, the left front being exactly the same as the right back, and *vice*

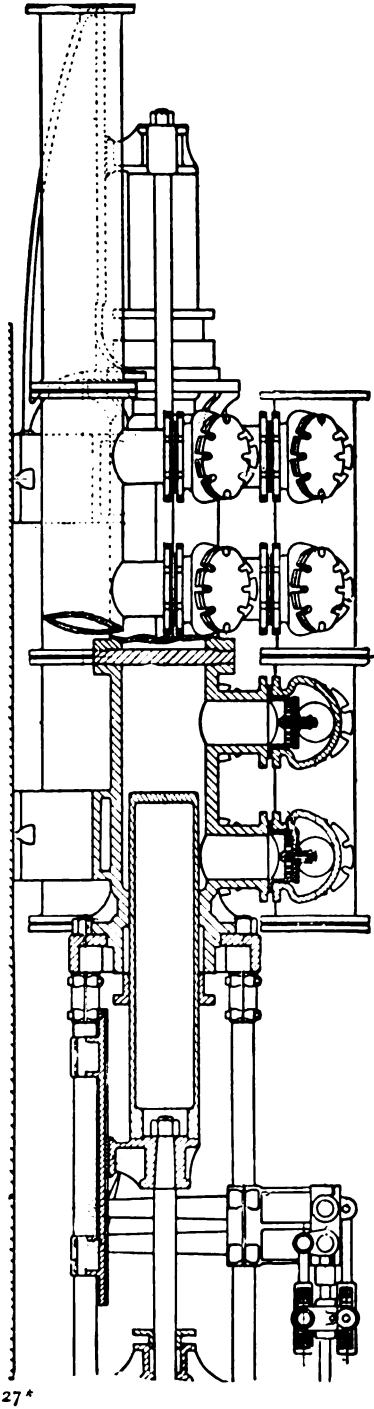


FIG. 217.

versa. This arrangement also serves to preserve the alignment of pump, as this tie-rod flange, as it is called, being a separate casting from the working-barrel, and not exposed to the action of the acid water, will last as long as the steam-end, so that in renewing a front or inside working-barrel it is only necessary to place it in position and draw up the bolts to insure the pump being in perfect line. To still further facilitate the renewal of inside working-barrels, the tie-rods are turned to the same size as that portion which goes through the bosses on the tie-rod flange; cast-iron split clamps or collars, shown in the drawing, are bolted thereon of sufficient length to allow of the tie-rod flange being drawn towards the steam-end far enough to clear the working-barrel, which can then be removed by lateral movement without disturbing the steam-end.

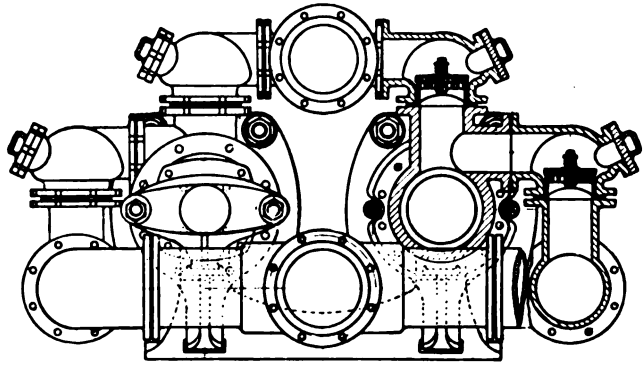
The pump has four single-acting, outside-packed plungers connected in pairs by two steel parallel rods coupled to strong cast-iron cross-heads, which, in turn, are fastened to plungers by T-headed bolts let into cored recesses in the plungers. The plungers are supported on the outboard end by adjustable shoes or gibs on the bottom of each cross-head. These run in slides, as shown. The writer considers this outside form of connection the best, even for very moderate lifts, in mine-work, as it does away with the necessity of bronze rods and connections, which are always a source of more or less trouble and uncertainty when used in acid or gritty water. The plungers are made of hard, close-grained iron, which will, if properly packed and lubricated, last fully as long as brass ones. The plunger gland-bolts have T-heads, and are let into cored recesses on the outside of the stuffing-box, the same as the bolts securing the plungers to the cross-heads. There are no studs whatever in the water-end.

The water-end has 16 valve-chambers with a single valve in each. A sectional elevation of the valve-chamber is shown in the longitudinal section, Fig. 217, and in the end elevation, Fig. 218.

The use of a single large valve to each chamber instead of

a number of small ones, Mr. Rood thinks, has many advantages when applied to mine-work. It may be well to state that the mine water in the anthracite coal-fields not only attacks iron but brass and phosphor-bronze as well, so that the life of a valve-seating is at best a limited one. This fact alone makes it imperative that pump-valves for mines be simple, strong, easily examined, and quickly replaced. It must be admitted that 8 or at the most 16 large valves are

FIG. 218.

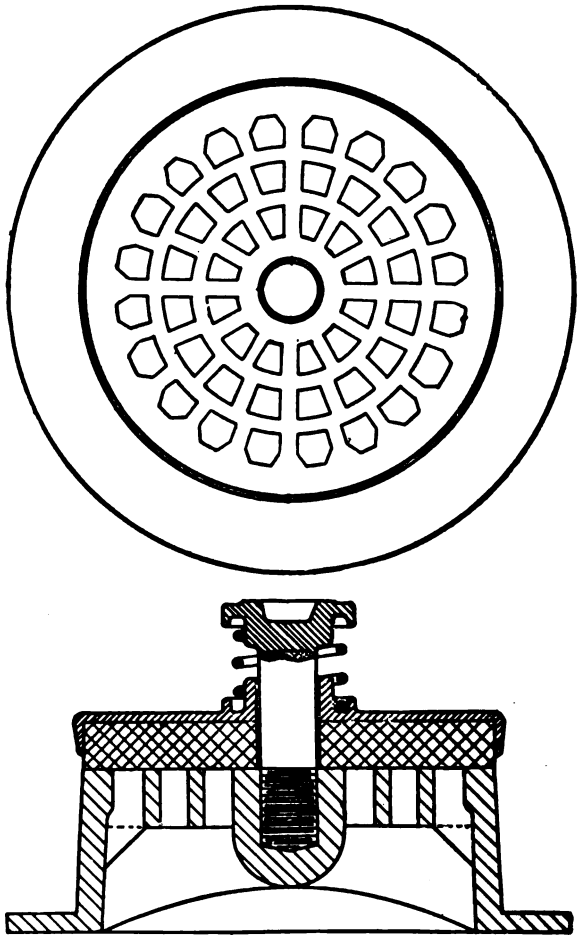


much easier kept in repair, and can be removed much more rapidly than from 50 to 100 small ones. The several details, such as valve-stems, springs, etc., are always much larger and stronger for large valves, and are better able to withstand the over-effort of an energetic man pulling on a long wrench; those who have had small valve-stems twisted off in this manner with the water gaining on the pumps will fully appreciate the importance of this detail. An enlarged illustration of the valve and its seat is shown in plan and sectional elevation in Fig. 219. The valve-seat is held in place by a flanged joint between the valve-chamber and the pump or pipe-flange underneath. The valve is made of india-rubber with a gun-metal cap; a phosphor-bronze spring and bolt completes the arrangement as shown.

While 8 chambers and valves would have answered as well as the 16 small ones, so far as the proper and smooth

working of pump is concerned, the desirability of keeping the chambers as small as possible for ease in handling in case of renewal, etc., determined in this case the use of the smaller pattern.

FIG. 219.



This form of valve-chamber, while differing radically from the usual style, has after a thorough trial proved itself so well fitted for the work as regards durability, etc., that they have

adopted it as a standard for their larger permanent station mine-pumps, as combining the greatest strength with the least weight of metal, the largest clearance around the periphery of the valve without any waste room elsewhere, and finally as costing very little to fit up and consequently not expensive to renew.

The question of room around the periphery of a valve is an important one in mine-pumps, as the water, in most cases being acid and carrying large quantities of fine coal and grit, if allowed to impinge against the walls of a chamber as it rushes out from under the valve, will cut the metal away so rapidly as to ruin the chamber in a short time. In the chamber illustrated it will be noticed that in addition to giving an extra amount of room at this point, its shape is such as to turn the course of the water upwards and inwards instead of against the walls. It is to this feature that is attributed the long life of this style of chamber, which invariably outlasts all the other parts of the water-end.

The method of securing the seatings between flanges of a chamber and the water-cylinder, while not new, is particularly adapted to mine-pumps, as it admits of very easy and quick renewals, there being no fits to make in the chamber, and it has the further advantage that no leak past the seat can occur without being instantly detected and stopped.

Steam-Pipes for Collieries.—An excellent paper, prepared by Mr. E F. C. Davis, and included in Vol. XI., "Transactions of the American Society of Mechanical Engineers," is here reproduced:

"The most common and the cheapest method of carrying steam, taking the world at large, is probably through wrought-iron 'gas-pipe' joined by the taper-thread, screwed into sockets or ferules.

"This answers admirably for small pipes, and even for comparatively large pipes where the conditions are favorable for screwing up the joints, and where the threads are not subjected to any serious corrosive action. Many of the steam-

pipe lines in the anthracite coal-regions, however, run for great distances underground, through contracted slopes and headings where it is almost impossible to make the screwed joint. In the screwed socket-joint there is always some space between the ends of the pipes, and the condensed steam from the best available feed-water is so corrosive that a cutting or furrowing action takes place between the ends of the pipes and the ferule, which sooner or later causes leakage. It is then impossible to tighten up these screwed joints without screwing up the whole pipe-line.

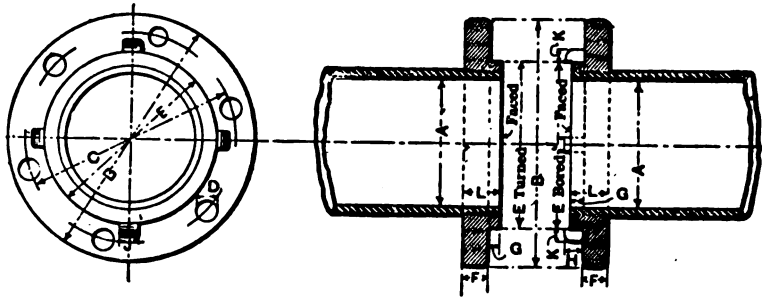
"Some of these difficulties are avoided by the use of 'flange-unions.' With these the pipe-line can be more conveniently put together underground, and in the event of a leaky thread the flange can be screwed on tighter, or a defective pipe can readily be replaced by a new one of the same length. But in the ordinary flange-union there is a space between the ends of the pipes, and the above-mentioned corrosive action is so destructive to the threads that cast-iron pipes have generally been considered necessary for reliable and durable steam-pipe lines; though the first cost is about double that of wrought-iron pipe.

"In view of the foregoing, the Philadelphia and Reading Coal and Iron Company has adopted the flange-joint shown in annexed cut for all colliery steam-pipes. These flanges are screwed tightly on the pipe,—the pipe carried in a steady rest,—and the end of pipe and flange faced off flush with each other. The lugs are at the same time bored out, and the projection turned off concentric with the bore of the pipe. This insures perfect continuity in the pipes, and the lugs also centre the gum-joint rings accurately, so that a gum-joint is obtained between the abutting ends of the wrought-iron pipes. The continuity of the bore of the pipe insures a free flow of steam and condensed water, so that all liability to furrowing at the joints is avoided and the gum-joint formed between the ends of the wrought-iron pipes protects the thread from all danger of corrosion. If an odd length of pipe needs to be made at a colliery, the pipe, if not over four inches, can be threaded

with a hand-stock and die, and a finished flange screwed on until the pipe projects through. The pipe must then be filed off flush with face of flange.

"In moulding these flanges it is best to have the pattern arranged to leave its own cores. This insures accu-

FIG. 220.



Size of Pipe.	Diameter of Flange.	No. of Holes.	℄ of Holes.	Size of Holes.	Diagram of Facing Piece.	Thickness of Flange.	Depth of Facing Piece.	No. of Lugs.	Length of Lug.	Width of Lug.	Thickness of Lug.	Total Thickness of Flange.	Length of Pipe.
A.	B.		C.	D.	E.	F.	G.		H.	J.	K.	L.	
													Feet.
3″	7¾″	4	6″	¾″	5″	¾″	5/16″	4	⅞″	¾″	5/16″	1⅛″	16
3½″	8½″	4	6½″	¾″	5½″	¾″	5/16″	4	⅞″	¾″	5/16″	1⅛″	16
4″	9½″	4	7½″	⅞″	6″	⅞″	5/16″	4	⅞″	¾″	5/16″	1⅜″	16
5″	10½″	4	8½″	⅞″	7″	⅞″	5/16″	4	⅞″	¾″	5/16″	1⅜″	16
6″	12″	6	10″	⅞″	8″	1″	5/16″	4	⅞″	¾″	½″	1½″	20
7″	13″	6	11″	⅞″	9″	1″	½″	4	⅞″	¾″	½″	1½″	20
8″	14″	6	12″	⅞″	10″	1¼″	½″	6	⅞″	1″	½″	1¾″	20
10″	16¼″	8	14″	1″	12″	1¼″	½″	6	⅞″	1″	⅝″	1¾″	20

Pipe must be screwed through flange so as to be steam-tight, and flange and end of pipe faced off flush at one operation.

racy in the positions of the bolt-holes and the large central hole, relative to each other and to the other parts of the flange.

"Several thousand feet of steam-pipe fitted with these flanges have been put in service, and have all proved perfectly satisfactory."

Water-pipe for mines should be of hard cast iron made in uniform lengths, so that in the event of having to replace a section of pipe it can be easily done, and without having to take the time necessary to face it off to an exact length. The pipes for the anthracite coal-mines of this State are generally cast with what corresponds to a chipping-piece in ordinary work, this projection being one-fourth of an inch above the body of the flange, by one and a half inches wide. This chipping-piece is cast against a cast-iron chill, thereby insuring a smooth and true face for making a joint. The chill is so arranged as to include the bolt-hole cores for the flanges, so that the pipes require no machine-work whatever. Fig. 221 represents a section of a joint made up. For column or delivery-pipes a wrought-iron welded ring ¼ × 1 inch iron is made with an inside diameter half an inch larger than the bore of the pipe; this ring is spirally wound with strips of cotton cloth 1½ inches wide, previously boiled in coal-tar until thoroughly saturated. For making slight bends in a pipe-line, a bevelled ring of either wrought or cast iron similarly wrapped as above is substituted for the ring having parallel sides. (See Fig 222.)

FIG. 221.

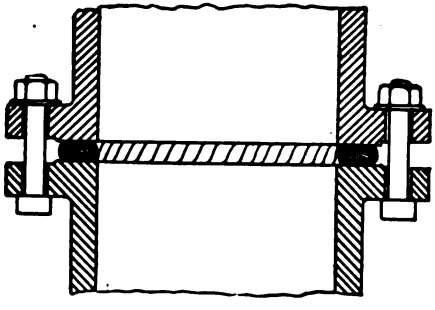
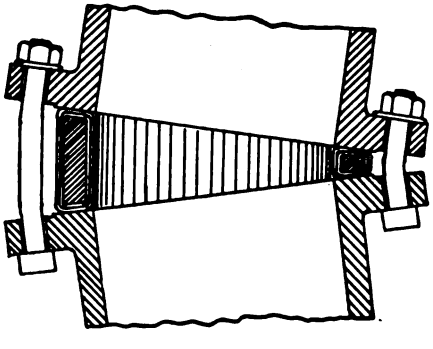


FIG. 222.

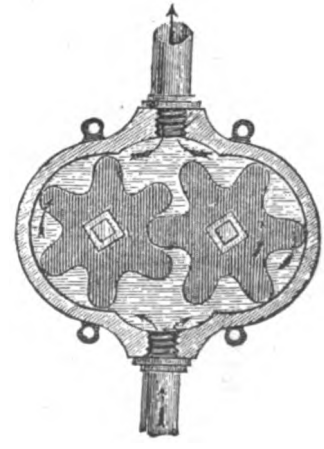


CHAPTER XV.

ROTARY PUMPS.

Rotary motions were, as Ewbank suggests, favorite ones with ancient philosophers; they considered a circle as the most perfect of all figures, and erroneously concluded that a body in motion would naturally revolve in one. In later times, as the problem of pump design began to press itself upon the attention of mechanics, it was a favorite one with many to convert the intermittent action of the pump-piston into a continuous circular movement, that the power expended in constantly bringing all the water in the cylinder and suction-pipe alternately to a state of rest and motion was saved, because the liquid would be kept in constant motion in passing through the pump.

FIG. 223.



One of the oldest and at the same time one of the best rotary pumps is shown in Fig. 223. No precise date can be assigned for its invention, but it existed in its present form probably early in the seventeenth century. Two cog-wheels, the teeth of which are fitted to work accurately into each other, are enclosed in an elliptical case. The sides of these wheels turn close to those of the case, so that water cannot enter between them. The axle of one of the wheels is con-

tinued through one side of the case, and the opening made tight by a stuffing-box or collar of leather. A crank is applied to the end to turn it, and as one wheel revolves it necessarily turns the other; the direction of their motions being indicated by arrows. The water that enters the lower part of the case is swept up the ends by each cog in rotation, and as it cannot return between the wheels in consequence of the cogs being there always in contact, it must necessarily rise in the ascending or forcing-pipe. The machine is therefore both a sucking and a forcing one, as indeed all rotary pumps are.

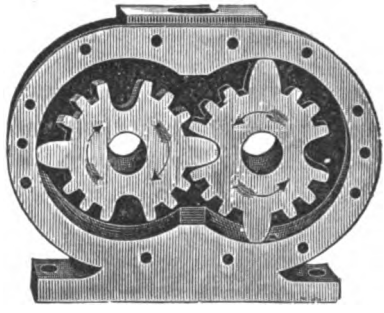
Rotary pumps may be divided into classes, according to the forms of and methods of working the pistons, or those parts that act as such, and according to the various modes by which the *butment* is obtained. It is this last that receives the force of the water when impelled forward by the piston; it also prevents the liquid from being swept by the latter entirely round the cylinder or exterior case, and compels it to enter the discharging-pipe. In these particulars consist all the essential differences in rotary pumps. In some the butments are movable pieces that are made to draw back to allow the piston to pass, when they are again protruded till its return; in others they are fixed and the pistons themselves give way. It is the same with the latter; they are sometimes permanently connected to the axles by which they are turned, and sometimes they are loose and drawn into recesses till the butments pass by.

An objection to rotary pumps is that they are noisy, wear out rapidly, and are costly to repair. They do not have a high efficiency, to begin with, and this grows constantly less with wear. They are, nevertheless, very useful pumps, and in the smaller sizes, at least, are used in large numbers throughout the country in manufacturing and mercantile establishments.

The engraving, Fig. 224, shows the internal arrangement of rotary pumps as made by Rumsey & Co., who claim that this form of revolving cam or piston has been demonstrated after many years of experiment and successful practice by the

most prominent mechanics and manufacturers to be the best in use, and of the proper shape to insure the very minimum of friction and wear.

FIG. 224.



The arrows shown in the cut indicate the direction in which the cams should be run,—*i.e.*, inwardly, or towards each other at the top or discharging orifice.

They have built and placed these pumps for many years, claiming they have given uniform and unequalled satisfaction in all their situations and uses, while comparatively cheaper and stronger than any other first-class rotary. They have heavy cast-steel mandrels, and babbit-metal boxes, with two pairs of heavy gears to relieve the cams. They discharge a large and constant stream without the aid of an air-chamber, can be run at a very high rate of speed without injury, and will operate for months at a time without cessation, having no packing or valves to become obstructed or worn out. There is nothing about these pumps but metal. They are simple, and all essential parts can be got at without trouble. They will not clog in pumping thick fluid or matter, and they are made non-freezing when not in use. The motion being rotary or circular, the strain upon the belt or gear is equal at all points. Their qualities well adapt them for establishments, including distilleries and breweries, for pumping hot liquors; in oil- and sugar refineries, paper-mills, and starch-factories; in slate-, stone-, and marble-mills, where they serve the double purpose of supply and freeing the quarries of muddy and gritty water; at railway-stations, mines, salt-wells, water-works, and also for wrecking-pumps, as they will not become clogged with wet grain, etc.

Rotary-pump design has engaged the attention of inventors for many years; unfortunately, most of them have

been more or less badly designed, and in consequence the mere use of the word "rotary" in connection with any pump seems at the present time a sort of disparagement which, according to M. Poillon, the principle does not merit.

On looking into the cause of this state of things, the mischief is readily perceived to have arisen from neglect, on the part of inventors and makers of rotary pumps, of some very simple facts in connection with the subject. In most cases the following points have practically been lost sight of:

1. That water is practically incompressible, and that in a liquid mass the pressure and the velocity at different points bear an invariable relation to each other.

2. That in any machine which has attained its regular working speed the driving power expended is equal to the sum of the useful work done plus the useless resistances overcome, and that, when proper care is not taken to prevent such a result, the useless work may absorb actually the greater part of the power.

3. That, whatever care be taken in the construction of a machine, two metal surfaces cannot be made to rub one upon the other in water without occasioning their rapid destruction by friction, but that if simply in contiguity without friction they will work without injury.

4. That, as the speed increases, the friction of the working parts in a pump very quickly becomes a matter of great importance and absorbs a considerable amount of work, while, on the contrary, absolute contact between parts in motion is a matter of no practical importance, the leakage becoming relatively less at higher speeds.

Hence may be explained the failures attending so many designs of rotary pumps, due equally to the want of appreciation of theory and to neglect of the teachings of practice.

An examination into the efficiency of many of the rotary pumps now in use show that it is very low. The great excess of driving-power over useful work done arises from two principal causes:

1. The inertia of the water, or difficulty of putting it into

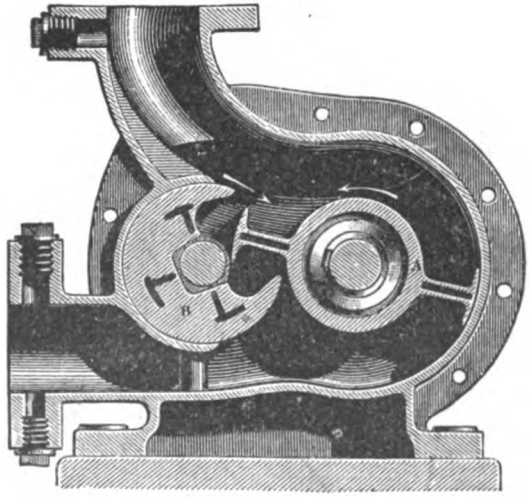
motion again after it has been brought to rest, and the consequent reduction of the effective pressure.

2. The necessity of imparting at certain moments a high velocity to a considerable mass of water, the production of this velocity requiring the expenditure of a great amount of power, of which only a small portion is given out again as useful effect.

It is clear that if these two sources of difficulty are got rid of, a near approach is made to perfect efficiency; that is, to an equality between the theoretical driving-power required and that which is utilized in the work done.

The Greindl pump, illustrated in Fig. 225, consists of a chamber within which work two cylindrical drums, *A* and *B*, of equal diameter, running in contact with each other on

FIG. 225.

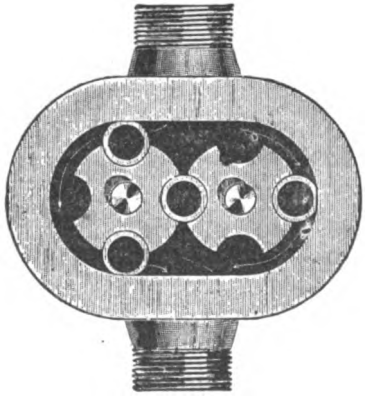


parallel shafts. One of these drums, *A*, carries two radial vanes or blades acting as pistons, which as they revolve enter alternately into a recess of epicycloidal section extending along the whole length of the other drum, *B*. The shafts of

the drums are geared so that the recessed drum, *B*, makes two revolutions to one of the bladed drum, *A*, thereby enabling the single recess in the quick drum to serve for the two blades on the slower. The inlet and outlet passages are arranged in such a manner as to present everywhere the same sectional area throughout the entire course of the water, in order not to impede its movement in any way during its passage through the pump. In consequence of the continuous motion of the stream of water, any foreign solid substance can pass through the pump without occasioning either a stoppage or a breakage. The blades of the slower drum strike the water without any perceptible shock. Lateral pockets in the end-cover plates afford ample space for the water to escape through at the moment when the space left between the blade and the recess threatens to be insufficient for that purpose.

In rotary pumps, by simply changing the speed of working, the actual quantity of water delivered can be varied within certain limits without producing any change in the proportionate efficiency of the pump, whatever its size, an advantage which cannot be obtained with centrifugal pumps.

FIG. 226.



The Berrenberg rotary pump is shown in sectional elevation in Fig. 226; each piston is a circle, from which four semicircular sections have been cut. In two of the semicircles, opposite each other, tubes are bolted through from the semicircular spaces left vacant. It is so constructed that when the pistons revolve, the tubes (or boxes, as they are called) of one piston fit into the vacant spaces of the other piston with perfect accuracy; and

the circumferences of the pistons being always in contact with each other, it will be seen that there is no time in the course of their revolution when there is much chance for leakage. In case of wear, it can be made good by removing the tubes (or boxes) and putting a few thicknesses of paper packing in the seat, or by turning the tubes a little so as to present a new surface to the point of contact with the case. The pump is double-geared, and has inside and outside bearings; these latter are tapered, so that any wear in the bearings or shaft may be taken up and the shafts kept true.

TABLE XIX.

EXPERIMENTS ON THE EFFICIENCY OF A 2-INCH BERRENBERG ROTARY PUMP, CONDUCTED BY R. A. HALE, HYDRAULIC ENGINEER, APRIL, 1888.

Number of Experiment.	Revolutions of the Pump per Minute.	Gallons pumped per Minute.	Height of Centre of Pressure-Gauge above Water in Pit. Feet.	Reading of Pressure-Gauge. Pounds.	Total Lift in Feet.	Depth of Water in Feet on Weir.	Cubic Feet of Water discharged per Second.	Horse-Power used in driving Pump.	Actual Work performed pumping against Lift. Horse-Power.	Efficiency of Pump. Ratio of Useful Effect to Power expended.
1	690	120	7.00	10.6	31.49	.233	.268	4.62	0.96	.207
2	722	127	7.03	25.2	65.36	.242	.284	6.05	2.11	.350
3	698	126	7.04	21.4	56.47	.241	.282	5.03	1.80	.359
4	697	124	7.05	31.5	79.81	.237	.275	5.81	2.49	.428
5	580	107	7.10	8.65	27.08	.215	.239	2.36	0.73	.311
6	593	104	7.10	24.0	62.54	.210	.231	3.43	1.64	.478
7	583	99	7.15	40.3	100.24	.203	.221	4.88	2.51	.515
8	573	93	7.20	55.15	134.60	.194	.206	6.47	3.14	.486
9	575	96	7.25	41.5	103.11	.200	.215	5.04	2.51	.499
10	591	101	7.33	27.2	70.16	.207	.226	3.71	1.80	.485
11	530	83	7.43	72.9	175.83	.180	.185	8.61	3.68	.427
12	478	86	7.35	6.3	21.90	.185	.192	1.76	0.48	.270
13	479	81	7.40	22.9	60.30	.177	.180	2.42	1 23	.510
14	468	76	7.40	33.1	83.86	.170	.170	3.14	1.62	.514
15	451	70	7.40	47.6	117.36	.161	.156	4.01	2.08	.518
16	462	69	7.40	65.2	158.00	.159	.154	5.98	2.76	.462
*17	475	81	7.40	23.5	61.70	.177	.181	2.43	1.27	.521

* 3-inch belt in this experiment.

The power was measured by a Webber dynamometer of 10-horse-power capacity. The pressures were measured by a gauge attached to the discharge-pipe, about seven feet above

the level of the water in the tank from which the water was taken.

The water was measured over a rectangular, sharp-crested weir, with two end contractions. The length of the weir was .763 feet, and the depth was measured by a hook-gauge, and quantity of water passing was measured by the formula of J. B. Francis, C.E.: $Q = 3.33\ (L - \frac{2}{10}\ H)\ H^{\frac{3}{2}}$, thus giving the quantity in cubic feet per second.

The pulley for the 3-inch belt proving too small to convey the power for the range of pressures which it was desired to run, the loose and tight pulleys were blocked together, and a 6-inch belt was used. The accompanying table comprises seventeen experiments, divided in three groups of varying speeds obtained by shifting the size of the pulleys on the counter-shaft; each group covers a range of various pressures, from which can be determined the pressures and speeds to give the best efficiency in the working of the pump. The first group was run at a much higher speed than the pump was intended to run, and a wide range of pressures was not taken.

The second and third groups were more nearly alike in efficiency; the third one, with the lowest speed, showing the best result.

The varying pressures were produced by partly closing the valve to the discharge-pipe, which, in general, decreased the amount of water pumped.

Pressures in Pounds per Square Inch.	Efficiency of Pump = Ratio of Useful Effect to Power expended.			
	Group 2.		Group 3.	
10.	.30		.30	
20.	.41		.45	
30.	.48	Mean,	.52	Mean,
40.	.50	.493	.515	.517
50.	.50		.515	
60.	.485		.50	
70.	.45		.455	
80.	.41			

It will be seen that, between 30 and 50 pounds pressure, the average efficiency is .493 with 575 revolutions of the pump, and .516 efficiency with 460 revolutions per minute, the efficiency falling off above and below that point.

The above may be taken as a fair sample of the performance of rotary pumps generally, and the few illustrations given will suffice our present purpose. To enter upon a description of the various designs for rotary pumps would require too much space in this publication; the writer suggests to those interested in rotary-pump design the perusal of Kennedy's translation of Reuleaux's "Kinematics of Machinery." The elementary details are systematically classified; sectional elevations are given by which any particular construction may be quickly identified.

CHAPTER XVI.

CENTRIFUGAL PUMPS.

A DEMAND exists for a pump with which large volumes of water may be quickly handled, such as in tanneries, paper-mills, print-works, dry-docks, etc. This water being liable to contain chips, bark, and other floating matter, makes it especially desirable that the pump be valveless,—a problem for which the centrifugal pump offers an almost complete solution. The construction of the centrifugal pump is exceedingly simple, consisting of a revolving fan having two or more blades, either straight or curved, attached to a revolving spindle, and fitted in a case or shell so constructed that the suction shall enter at the centre of the wheel, and the delivery placed tangent to the outer path of the revolving blades.

The recorded experiments relating to centrifugal pumps are few. Among these, two are favorably known to the writer,—one by Mr. R. C. Parsons,* England, and the other by Mr. W. O. Webber,† Lawrence, Mass,—the first relating more especially to the theory of centrifugal pumps, and the latter to their efficiency as compared with reciprocating pumps; the writer acknowledges his indebtedness to both of these experimenters for subject-matter used in this chapter.

Centrifugal pumps are by no means a modern invention, the crude idea of which probably dates as far back as the middle of the last century, when the mathematician

* Proceedings of the Institution of Civil Engineers. London, 1876. Vol. XLVII.
† Trans. Am. Soc. Mechanical Engineers. New York. Vols. VII. and IX.

Euler brought out a primitive form of centrifugal pump, an account of which he published in the Proceedings of the Academy of Berlin for 1754, but which never came into practical use. From that period many rotary and centrifugal pumps were invented, principally by French engineers, but none of them seemed to have yielded even a reasonably good efficiency. The first mention of a centrifugal pump at all to be compared with those of the present day is in the year 1830, when one was erected by Mr. McCarty in the navy-yard at New York, and some improvements were patented by him in the following year. The next epoch in the history of the centrifugal pump is the Exhibition of the year 1851, London, when the late Mr. J. G. Appold achieved a great success with his pump of trebling the efficiency obtained by any other exhibitor.

The experiments made with Appold's centrifugal pump showed that its efficiency mainly depended upon the form of the blades of the fan; and, further, that the best form was a curved blade pointing in the opposite direction to that in which the fan revolved. These were tried in comparison with two other forms.

TABLE XX.

APPOLD'S CENTRIFUGAL PUMP.

	Height of Lift in Feet.	Discharge in Gallons per Minute.	Revolutions per Minute.	Velocity of Circumference in Feet per Minute.	Percentage of Effect to Power.
With radial arms . . .	18.0	474	720	2262	24
With straight inclined arms	18.0	736	690	2168	43
With curved arms . . .	8.2	2100	828	2601	59
" " . . .	9.0	1664	620	1948	65
" " . . .	18.8	1164	792	2988	65
" " . . .	19.4	1236	788	2476	68
" " . . .	27.6	681	876	2751	46

From the above table it will be seen that as between radial blades and curved blades the increase in efficiency was more

than doubled in favor of the latter; subsequent calculations and more exhaustive experiments show that nothing in connection with centrifugal pumps has been more clearly proved, both theoretically and experimentally, than that the blades should curve backward, and according to Thomson the general form of the curved blade is an element of great importance, and it is a necessary consequence of the arms being thus shaped that the water is driven through the wheel, partly by the action of centrifugal force and partly by the oblique pressure of the blades on the water. The ratio which each of these forces bears to the other varies even in the same pump, according to the proportion the speed of the pump bears to the height of the lift. When the lift is very small and the speed great, the water is expelled without having any considerable rotary motion imparted to it. In such a case the resistance to the outward motion of the water is so small that the oblique action of the blades is sufficient to expel it without giving it a speed of rotation at all approaching that of the fan.

In centrifugal pumps the main object is to pass the water through the pump with as little "whirling" velocity as possible, because the power thus absorbed is not again given out, and becomes a more or less hinderance to the flow. Designs vary so much that it is clear no settled opinion has yet been reached as to the best proportions for blades, nor yet as to the best proportions and best shape for the case which contains the revolving fan. Mr. Thomson recognizes it as a favorite idea with many designers of centrifugal pumps to contract the space between the sides of the fan at the circumference so as to make the area of opening through the fan uniform, and thus preserve a uniform radial motion of the water during the whole passage through the fan. The effect of this as well as of many other modifications was tested experimentally by Mr. Appold, and he found that it was productive of no increase of duty. Mr. Parsons, on the other hand, considers it a detail which affects the efficiency of the pump, but not to the same extent as the form of the blades, and says that "the

old theory which Morin, Appold, and many others held, and which is still held in some recent books,—viz., that as long as the casing outside the fan is large enough it is immaterial what shape it is,—can be proved to be false both by theory and experiment."

The passages throughout the pump must be so proportioned as to have a gradually increasing velocity in the water until it arrives at the circumference of the fan, and then to have a gradually decreasing velocity until it issues from the discharge-pipe. This condition is effected by having a conical end to the suction-pipe, and, what is much more important, is to have a spiral casing surrounding the fan. The importance of this last detail, Mr. Parsons says, is shown most conclusively by experiments with both circular and spiral cases. The form of the casing should be an Archimedian spiral, which has the property that the water flowing round the case moves with the same velocity as that issuing from the fan. The casing should then gradually open out into the discharge-pipe.

TABLE XXI.

EXPERIMENTAL EFFICIENCIES OBTAINED BY R. C. PARSONS ON A 14-INCH REVOLVING FAN, 10-INCH SUCTION, AND 10-INCH DISCHARGE, MADE ON THE APPOLD PRINCIPLE.

Revolutions per Minute.	Gallons per Minute.	Lift in Feet.	Foot-Pounds.		Efficiencies. Per cent.
			Water Raised.	Indicated Power.	
392	1012	14.67	148,461	298,438	49.74
394	1108	14.70	162,875	317,158	51.35
395	1197	14.65	175,364	332,136	52.80
400	1431	14.75	211,073	374,954	56.20
405	1695	14.75	251,987	419,790	60.17
425	1108	17.20	190,576	388,316	48.97
431	1431	17.40	248,994	447,552	53.63
435	1695	17.60	298,310	486 050	61.37

This pump was placed on a floating scow to obtain as nearly as possible a constant lift; it was driven by a separate steam-engine, and the power was measured by a dynamometer.

There are two totally different conditions in which a cen-

trifugal pump may be situated while it is rotating,—one in which it is revolving just fast enough to raise the water up to the discharge-pipe and no farther, and another in which it is revolving slightly faster, and is discharging water out of this pipe. In the first case there is only centrifugal force, which is produced by the water in the fan rotating, that maintains the column of water in the discharge-pipe. In the second case, this force is still produced, but in addition to it another, which may be called the force of impact, or in other words, the force with which the blades of the fan impinge against the water discharged by the pump.

The centrifugal force in the first case was calculated by Mr. Parsons to be as follows: Assuming that the fan is a cylinder of water; every particle of this water as it rotates exerts a force outwards from the centre; consequently the force exerted at the circumference, or that which maintains the head in the discharge-pipe, is the sum of the forces of all the particles from the centre of the fan to the circumference. This force is given in pounds per square inch by the formula

$$F = \int_0^R \frac{P x \, dx}{g} w^2 \qquad (1)$$

Integrating this expression

$$F = \frac{p R^2 w^2}{2g} \qquad (2)$$

p = weight in pounds of a column of water 1 inch square in section and 1 foot long.

R = radius of fan in feet.

w = angular velocity of fan.

g = dynamical force of gravity.

Now, since $R w = v$, where v equals velocity of circumference of fan, by replacing $R^2 w^2$ by v^2 in equation 2 it becomes

$$F = \frac{p v^2}{2g} \qquad (3)$$

Now, supposing that the head supported by the fan, while it is rotating with a tangential velocity v, be h, then the pressure at the base of the column is ph, but by the ordinary formula of dynamics

$$h = \frac{v^2}{2g}; \text{ therefore } ph = \frac{p v^2}{2g} \qquad (4)$$

thus by equations 3 and 4 $F = ph$.

Therefore a fan, when rotating, will support a column of water the velocity due to whose height is equal to the tangential velocity of the circumference of the fan. This conclusion is fully borne out by experiments, where corrections are made for the axle of the pump displacing a small quantity of water, and thus reducing to a slight extent the centrifugal force.

The Second Force exerted in this Case is that of Impact.—It is estimated by the maximum tangential velocity generated in the water passing through the fan, which takes place just as it is escaping at the circumference. The reason advanced that this force can be estimated by the tangential velocity produced is that no other force can produce this velocity.

The centrifugal force can only produce a radial force or radial velocity, but can in no case produce a tangential force or tangential velocity. This latter force can only be made use of by gradually reducing the velocity of the water issuing from the fan, and this condition is effected by the spiral casing and conical discharge-pipe, which can easily be calculated by multiplying v by cosine θ', the angle made by the blade of the fan at its outer extremity with the tangent to the fan; and subtracting this from V, the velocity of the circumference, the absolute tangential velocity of the water leaving the fan is obtained,—viz.:

$$v' = V' - v \cos \theta \qquad (5)$$

The head then due to this velocity is given by the formula

$$H_2 = \frac{v'^2}{2g} \qquad (6)$$

This, in other words, is the height that the water would rise supposing that there was no friction to impede it. Now, the circumferential force has been estimated in pounds per square inch; but by dividing it by 0.434 it is reduced to feet head of water. Then, by adding these two heads together, the theoretical height to which the pump lifts the water is obtained, —*i.e.*,—

$$H + H_2 = \frac{F}{.434} + \frac{v'^2}{2g} \qquad (7)$$

This theoretical lift is always greater than that deduced by experiment, and it is only in a perfect pump that these two lifts would coincide. Consequently, if the practical lift be divided by the theoretical lift, and the result multiplied by 100, the percentage efficiency of the pump is obtained.

Effect of High Speed of Rotation.—The faster the fan rotates, the lift remaining constant, the smaller is the centrifugal force. This seems to be a paradox at first sight, but the reason is evident. As the discharge increases, the velocity of the water in the casing more nearly approaches that of the water leaving the fan ; consequently the efficiency of the pump improves, and the theoretical lift diminishes, and with it the centrifugal force.

A remarkable property of centrifugal pumps may be mentioned, which has been clearly shown by experiment, and that is, a small increase in the number of revolutions of the pump, when it has begun to discharge, produces a very large increase in the delivery. Thus, in Table XXI. the difference in discharge between experiments at 392 revolutions per minute and 405 revolutions per minute is 683 gallons per minute, and this with the small increase of only 13 revolutions.

In the table of the Appold centrifugal pumps the highest efficiency given is 68 per cent., and in the table of efficiencies resulting from Mr. Parsons's experiments the highest efficiency is given at 61.37 per cent., which is practically the same as that given by Mr. Webber in his reference to the Gwinne pump, tested in 1883, under 14.7 feet lift.

Experimental Tests of Centrifugal Pumps by Mr. W. O. Webber.—Mr. Webber's use of the term efficiency he explains as indicating the value of $\frac{\text{Water H. P.}}{\text{Indicated H. P.}}$ for such pumps as are driven by an engine direct, and does not, therefore, show the full efficiency of the pump, but that of the combined pump and engine. It is, however, a very simple way of showing the relative values of different kinds of pumping

engines having their motive-power forming a part of the plant. Mr. Webber's tests were made with ordinary centrifugal pumps,

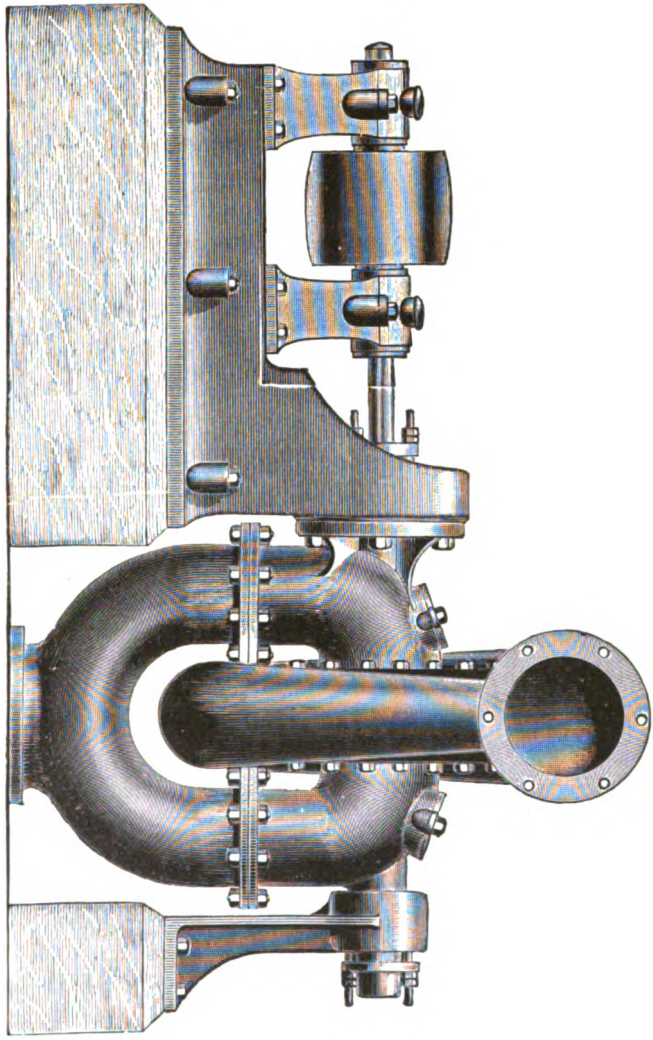


FIG. 227.

such as are supplied to the trade in the regular course of business. The general elevation of a 5-inch, class B pump is given in Fig. 227, and in sectional elevation in Fig. 228. In

calculating the efficiency of the pump, the cubic feet of water passing over the weir, measured by the hook-gauge, being converted into pounds by multiplying by 62.5, is again multi-

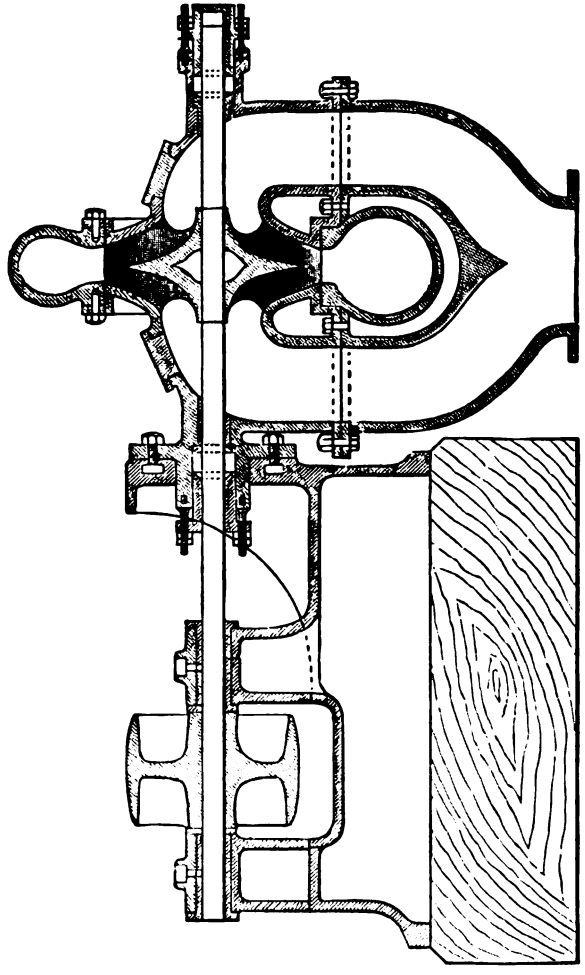


FIG. 228.

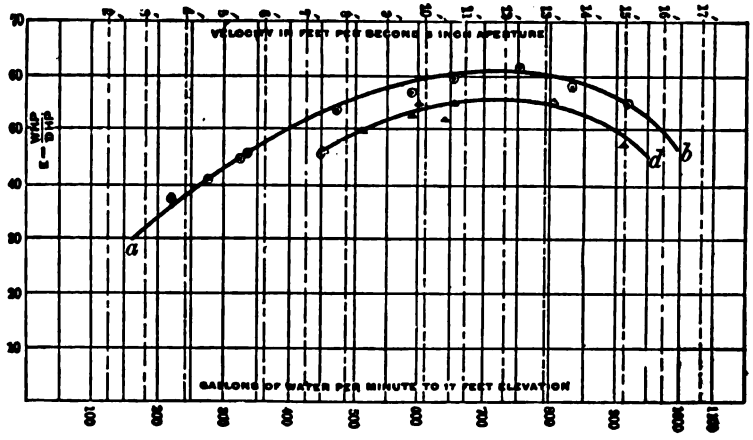
plied by the height from the level of water in the tank when the pump is running to the centre of the discharge-pipe, and the foot-pound so obtained, divided by 33,000, equals the water horse-power being developed.

The power used to do this work is measured by the dynamometer and equals the dynamometer horse-power; the water horse-power being divided by the dynamometer horse-power equals the efficiency of the pump being tested; or to formulate:

$$\frac{\text{Water H. P.}}{\text{Dynamometer H. P.}} = \text{the efficiency of such pumps as are driven by a belt.}$$

The diagram, Fig. 229, shows the efficiency curves for different velocities, plotted from tests made of two pumps with

FIG. 229.



5-inch apertures known on makers' lists as a No. 5, class B pump, and here illustrated.

These tests were made under an average elevation of 17 feet, the pumps in both cases draughting about 9 feet and discharging 8 feet higher. The upper curve *a b* was the result of tests made by a pump that was very clean and smooth inside. The lower curve *c d* was made by a pump in which, through carelessness in the foundry, the core sand had been allowed to burn into the inside face of volute or casing and water-passages. The difference between these two curves (which, by the way, are remarkably uniform) shows the absolute necessity of having the inside of all such pump-castings very

smooth and free from the slightest roughness. Both of these pumps were taken at random from stock, and were in no wise especially prepared for these tests. These tests seem to show that the efficiency rises very gradually and uniformly until the water reaches a velocity equal to 11½ feet per second. The highest efficiency with this size pump seems to equal a velocity of 12 feet per second, after which point the efficiency falls very rapidly.

A selection covering a considerable range of efficiencies is taken from Mr. Webber's table of data, as follows:

TABLE XXII.

TESTS OF CENTRIFUGAL PUMPS (NO. 5, CLASS B)—WEBBER.

Revolutions of pump . . .	390	410	445	460	470	510	585	590
Height of lift	16′8″	14′5″	17′10″	17′2″	17′2″	17′2″	17′10″	17′2″
Net load on dynamo . . .	14,850	18,200	23,200	25,905	28,036	34,188	32,790	40,000
Water cubic feet per minute	29.78	43.35	63.51	87.14	100.42	123.0	111.83	122.37
Water in gallons per minute	224.4	324.3	475.1	651.8	749.4	920.6	840.4	917.8
Water in foot-pounds . . .	31,294	39,181	70,585	93,458	107,250	136,731	124,820	131,295
Water horse-power	.948	1.19	2.14	2.83	3.25	4.08	3.78	3.98
H. P. for dynamo	2.7	2.4	4.0	4.7	5.3	7.4	6.5	8.6
Efficiency . . W. H. P. / D. H. P.	.35	.43	.53	.60	.61	.55	.59	.47

In another series of tests undertaken by Mr. Webber the results seemed to show that the efficiency of centrifugal pumps increases as the size of the pump increases, and which might be approximately stated as follows: that a 2-inch pump (this designation meaning always the size of discharge outlet in inches of diameter) giving an efficiency of 38 per cent., a 3-inch pump giving 45 per cent., and a 4-inch pump giving 52 per cent., were giving as good results, proportionally, as a 5-inch pump at 60 per cent., and a 6-inch pump at 64 per cent. of efficiency.

Table XXIII. gives a list of sizes of centrifugal pumps as made by the Lawrence Machine Company, together with the deliveries at various heights and revolutions per minute.

TABLE XXIII.

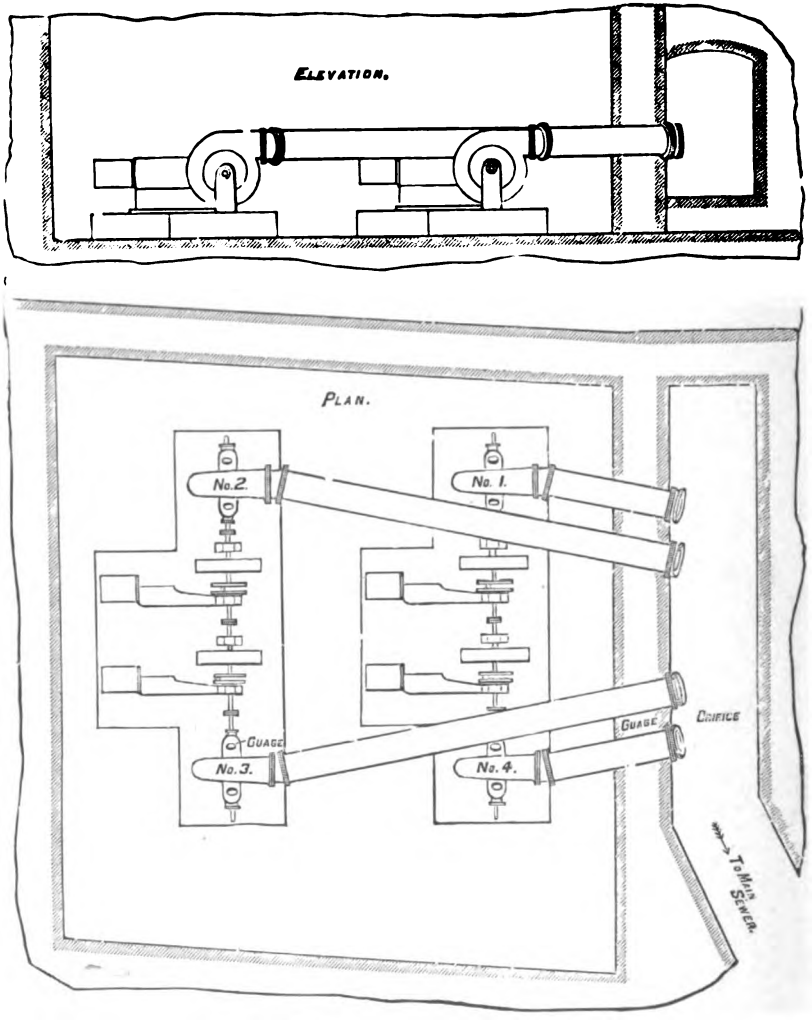
LAWRENCE MACHINE COMPANY.

Centrifugal Pumps for Lifts up to 35 Feet, known on Maker's List as Class "B." Table of Revolutions per Minute necessary to raise Minimum Quantity of Water to Different Heights with Different Sizes of Pumps.

Size of Pipes.		Economical Capacity in Gallons per Minute.	Horse-Power per Foot-Lift for Smaller Quantity.	Horse-Power per Foot-Lift for Larger Quantity.	Diameter of Pulley.	Width of Pulley.	Width of Belt.	Height in Feet and Revolutions per Minute.									Minimum Quantity of Water.
Suction.	Discharge.							6′.	8′.	10′.	12′.	16′.	20′	25′.	30′.	35′.	
2″	1½″	20 to 50	.024	.03	5	5	3	465	515	560	605	680	745	820	885	945	20
2½	2	60 to 80	.035	.05	5	5	4	425	475	515	560	625	680	750	810	870	60
3½	3	80 to 160	.055	.08	7½	7	6	390	435	475	510	575	630	695	750	800	80
4½	4	160 to 350	.075	.176	7½	7	7	365	405	445	475	535	590	645	700	745	160
6	5	330 to 600	.175	.30	12	11	8	320	355	390	415	470	520	570	610	650	330
6	6	500 to 900	.22	.378	14	11	9	285	315	345	370	415	460	500	540	580	500
8	8	1,100 to 2,000	.45	.84	16	12	10	215	240	260	280	310	340	375	410	435	1,100
10	10	1,600 to 3,000	.62	1.40	18	12	10	170	190	210	225	250	275	300	325	350	1,600
12	12	2,000 to 4,000	1.00	1.90	22	14	12	150	165	185	195	220	240	265	285	310	2,000
14	14	3,000 to 5,000	1.25	2.10	24	14	12	135	150	165	175	195	215	240	255	275	3,000
15	15	3,500 to 7,000	1.40	2.94	28	15	14	125	145	155	165	190	210	231	245	260	3,500
18	18	6,000 to 11,000	2.40	4.70	28	16	14	110	120	130	135	160	175	190	250	220	6,000
24	24	8,500 to 18,000	3.60	7.60													8,500
30	30	12,000 to 25,000	4.95	10.50													12,000

Efficiency test of a 24-inch centrifugal pump at the sewage station, Montreal. This test, conducted by George

FIG. 230.



H. Barrus, was with one of four centrifugal pumps made by the Lawrence Machine Company. The general arrangement of the pumps and engines is shown in Fig. 230.

The pumps are arranged in pairs, two being attached to one shaft, and two to another shaft. A pair of engines, with cranks 90 degrees apart, is placed in a central position between the two pumps on each shaft, the whole plant thus embracing four engines. The line-shaft for the two pumps and two engines is continuous, and the various sections are joined together by means of flange-couplings with recessed joints.

Each pump is fitted with a steam-ejector, by means of which the pump is primed, and the outlet end of the discharge-pipe is provided with a plain flap-valve, made of cast iron. The opening through the valve is the full size of the pipe. Each discharge-pipe is 27 inches in diameter, although the outlet of the pump is only 24 inches in diameter, and the pipe is made of wrought-iron plates, with riveted joints. As seen in the sketch, the water, on leaving the pump, is deflected from a straight course, a short bend being introduced connecting the pump to the discharge-pipe. With this exception, the pipe is straight.

The test was made on No. 3 pump. It embraced an indicator test of the quantity of power used, and a capacity test of the quantity of sewage discharged.

The pump was driven by the two engines on its shaft, and the No. 2 pump was allowed to run without being supplied with sewage. The character of the coupling between the two engines made it difficult to separate one engine from the other, and it was impracticable, for this reason, to make the test on a single engine and pump; but as the object of the test was to find the quantity of power expended upon the sewage, the use of the two engines and the supply of power for the friction of the whole shaft did not prove objectionable.

To determine the power expended upon the sewage, indicator diagrams were taken from each engine when the No. 3 pump was at work in a normal manner, and again when no sewage was being discharged, the pump running empty. The difference in the quantities obtained is taken for the desired result.

To determine the quantity of sewage discharged, an orifice was provided at the outlet end of the discharge-pipe, and the necessary data were observed for obtaining the result by calculation. The data used for the purpose were the head of water back of the orifice, shown by a mercury gauge, and the known velocity of discharge from an orifice of this kind under the observed head.

The orifice consisted of a ¼-inch iron plate, with an opening 20 inches in diameter, which was introduced in the flange-joint, between the flap-valve and the main discharge-pipe. The edges of the orifice were at right angles to the surface of the plate.

The nature of the sewage, according to a sample dipped up from the pump-basin, was but little different, in the matter of consistency, from that of pure water, although it was considerably discolored. All the solid material is removed before its entrance to the basins. The weight of a gallon of the sewage was not determined, but for the purpose of calculation, it has been assumed to be 8.4 pounds.

The corrected indications of the gauges averaged:

	Feet.
Mercury gauge. Actual head measured above centre of orifice	4.41
Suction gauge. Suction-lift, including all corrections	15.59
The total lift	20.00

Average revolutions of pump per minute, 157.8.
Boiler pressure varied from 75 to 91 pounds per square inch.

TABLE XXIV.

POWER TESTS.

	No. 3 Pump in Operation.	No. 3 Pump Running Empty.
1. Mean effective pressure, right-hand engine, in pounds	20.52	7.41
2. Ditto left-hand engine, in pounds	19.50	1.16
3. Ditto both cylinders, average, in pounds . .	20.01	3.12
4. Revolutions per minute	157 40	161.0
5. Indicated horse-power	126.18	20.16

The difference between the results of the two tests, which is 126.18 — 20.16 = 106.02, is the quantity of power expended upon the sewage.

The quantity of sewage discharged through the orifice, expressed in cubic feet per second, equals V, the velocity of the current in feet per second, multiplied by a, the area of the orifice in square feet, multiplied by c, the coefficient of discharge. The first quantity in this equation, V, is determined by the formula, $V = 8 \times \sqrt{\frac{h}{1 - \left(\frac{a}{A}\right)^2}}$, in which h is the head in feet, measured from the centre of the orifice, a the area of orifice, and A the area of the discharge-pipe. The quantity c, for orifices giving a perfect contraction, is, according to the best authorities, 0.6. The contraction in this case was not perfect, and the coefficient is assumed to be 0.65.

Substituting in the formula for velocity, $h = 4.41$ feet, $a =$ area of a circle 20 inches in diameter $= 314.16$, and A, area of a circle 27 inches in diameter $= 572.55$, the velocity works out 20.8 feet per second. The quantity discharged per second is, therefore, area of orifice (2.18 square feet) $\times$ velocity (20.8) $\times$ coefficient of discharge (0.65) $=$ 28.45 cubic feet per second. This corresponds to 212.8 gallons per second, or 12,768 gallons per minute.

The quantity of work performed, which is represented by the raising of this quantity of sewage through the range of lift noted,—viz., 20 feet,—is 12,768 gallons $\times$ 8.4 (pounds per gallon) $20 =$ 2,145,024 foot-pounds per minute. Reduced to horse-power, this becomes $\frac{2,145,024}{33,000} = 65.01$ horse-power.

The efficiency of the pump can now be calculated; that is, the relation borne by what may be called the useful work of the pump to the power expended upon it by the motor. The useful work, as noted, is 65.01 horse-power, and the horse-power expended by the engine is 106.02. The percentage of efficiency, is, therefore, $\frac{65.01}{106.02} \times 100 = 61.3$ per cent.

Efficiency of Large Centrifugal Pumps.—There is published in *Engineering*, Vol. LII., an abstract of a report by

Mr. A. Elink Sterk, Haarlem, on the efficiency of centrifugal pumps tested by him, October, 23, 1891, for the "Groote Ypolder," Holland. These pumps were constructed by Messrs. J. & H. Gwinne, London, but the dimensions are not given in the report, only the record of efficiency, which averages as follows:

Water horse-power	104.8
Coal consumption per hour per water horse-power	3.89
Steam per pound of coal	7.59
Steam per water horse power per hour	29.7
Efficiency $\frac{\text{W. H. P.}}{\text{I. H. P.}}$	0.656

The test lasted 4 hours and 2 minutes, the pump making in all 25,754 revolutions, or an average of 106.5 per minute. The steam-pressure varied from 93 to 96½ pounds. The mean discharge of water in cubic metres per minute was 116.7. The mean water horse-power was 104.8, the indicated horse-power being 159.8; therefore, $104.8 \div 159.8 = 0.656$ efficiency.

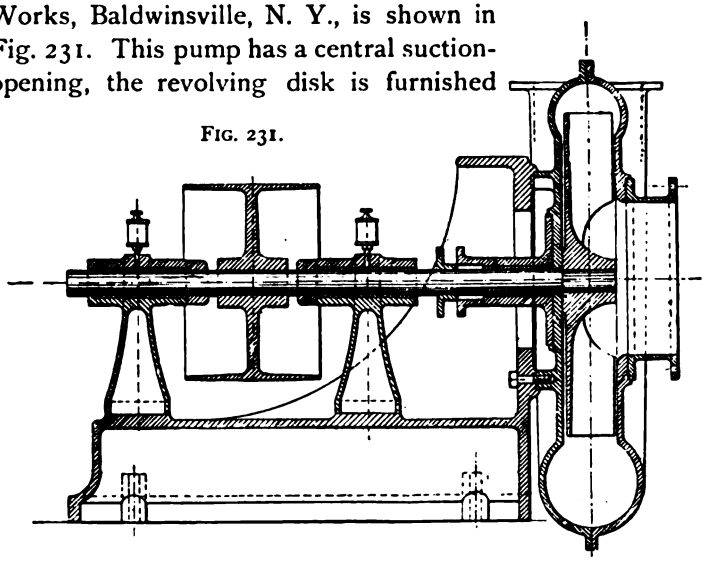
The coal consumption per hour and per water horse-power was, according to contract, to be 4.97 pounds; but it was actually found to be 3.89 pounds, or 1.075 pounds less than the contract figure.

The efficiency, or the relation of water horse-power and indicated horse-power, was for the whole duration of the trial a mean of 0.656, but, as appears now, the engine did not work during the whole trial at the most advantageous speed. If it had run at its best speed the efficiency would certainly not have fallen below 0.67, or fully 2 per cent. higher.

This high efficiency, Mr. Sterk says, is higher than that of any centrifugal pumping engine known to him. The next in this respect are the large centrifugal pumps at Hatakbeh, in Egypt, made by Farcot, at Paris, each of which is capable of raising 400 cubic metres per minute. The efficiency of these pumps was, according to trials made by Mr. Brühl, 0.651, which figure Mr. Sterk has no hesitation in saying is flattered.

Morris Machine-Works Centrifugal Pump.—A sectional elevation of a centrifugal pump by the Morris Machine-Works, Baldwinsville, N. Y., is shown in Fig. 231. This pump has a central suction-opening, the revolving disk is furnished

FIG. 231.



with curved blades for imparting to the water a proper direction and velocity into the spiral chamber which surrounds

FIG. 232.

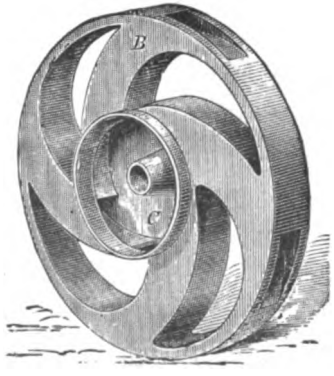


FIG. 233.

the circumference of the revolving blades. The makers have adopted this design in common with other first-class builders,

as it was experimentally shown long ago that spiral castings closely surrounding the fan gave the best results.

Several designs for pistons, as these revolving blades are sometimes called, have been tested in actual service, from which three have been selected for illustration.

Fig. 232 represents a hollow arm piston, which is used in their standard pumps in size No. 4 and above. It is the one on which the fame of the Heald & Sisco pump is mainly based, and is their special favorite for raising water or any thin fluid not too much encumbered with stringy or tenacious matter. For over a quarter of a century it has held the lead against all comers.

FIG. 234.

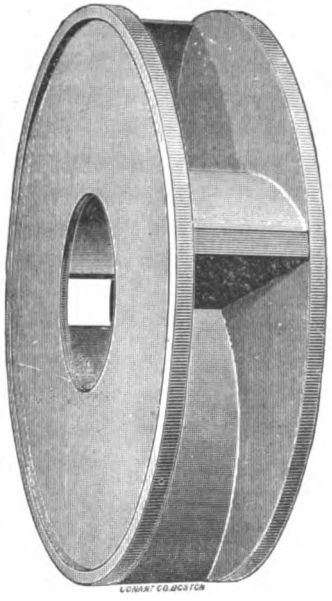


Fig. 233 represents a concave arm wing, which they use in their No. 3 standard pump and smaller sizes. It has proved itself very efficient. For raising half stuff in paper-mills and stringy material often found in tan liquor, they recommend the wing as being the best. In pumping very thick material they use a wing with two arms only.

The piston shown in Fig. 234 is of their own invention, and has been used upwards of six years in their special sand- and dredging-pumps. It is very heavy, has large openings, and is very efficient. By its use material taken in it is deposited on pump-scroll, thereby preventing wear on the pump-sides. All sand- and dredging-pumps of their make are fitted with it.

TABLE XXVI.

MORRIS MACHINE-WORKS—STANDARD HORIZONTAL CENTRIFUGAL PUMP.

No.	Economical Capacity in Gallons, per Minute.	Actual Capacity in Gallons, per Minute.	Horse-Power required for each Foot of Lift. Minimum Quantity.	Diameter and Face of Pulleys, in Inches.	Floor Space required, in Inches.
1½	20 to 40	160	.01	5 × 5	19 × 30
1¾	40 to 60	225	.016	6 × 6	21 × 33
2	60 to 80	325	.019	7 × 6	23 × 37
2½	80 to 100	400	.039	7 × 6	24 × 38
3	120 to 180	675	.047	7 × 7	25 × 39
4	200 to 300	1,300	.078	8 × 10	30 × 40
5	350 to 500	1,900	.14	10 × 10	34 × 54
6	500 to 700	2,700	.22	12 × 12	37 × 55
8	900 to 1,300	4,800	.34	18 × 12	45 × 63
10	1,600 to 2,200	7,500	.64	20 × 12	51 × 71
12	2,000 to 3,000	10,500	.88	24 × 14	62 × 78
15	3,000 to 5,000	16,500	1.20	30 × 16	77 × 80
18	5,000 to 7,000	22,000	1.80	36 × 18	93 × 103
22	7,000 to 10,000	35,000	2.90	48 × 18	126 × 130

The number of the pump is also the diameter of the discharge opening in inches. Where more than 25 feet of discharge-pipe is attached to the pump, they recommend using one or two sizes larger than the pump-discharge.

The horizontal right-hand pump is the one preferred by manufacturers generally. To use it without a priming apparatus the water must flow to the pump and rise as high as the top of the shell. They are the favorite for drainage and irrigation work, and when so used are generally fitted with a foot-valve. In point of execution there is no difference between the horizontal and vertical pump. The former has the advantage of being more readily examined in case of accident. All of their pumps are made to run right-handed, unless otherwise ordered. In their horizontal pumps the discharge, as usually made, will be in a vertical position, as shown in Fig. 235, unless another delivery is desired.

Double suction-pumps are to be preferred to standard or side suction-pumps, where there is a suction-lift of several feet;

but they can be recommended only for raising clear water, whereas the standard pumps will pass anything that will enter the suction.

Combined Pump and Engine.—When a pump cannot be conveniently driven by a belt on account of the situation of shafting, distance from power, etc., a combined pump and engine may be quite satisfactorily employed, especially for the larger sizes. Such a combination is shown in Fig. 235, which is not only very compact, but economical in the use of steam, making them especially desirable for use on shipboard, circulating pumps for ice-factories, etc.

For dredging operations a centrifugal pump fitted with a piston as shown in Fig. 234, has been found to give a high efficiency, the makers claiming that it will wear longer and cost less for repairs than the forms heretofore in use. These pistons are very heavy, twice as much as in ordinary pumps, and are fitted with a water-bearing that will prevent the sand from cutting out the journal and the shaft, thereby prolonging the life of the pump at least double over one fitted with a piston similar to Fig. 233; the latter being good for water pumping, has proven itself extravagant in the use of power, but wanting in durability on account of the rapid wear, because the sand is dragged at all times between it and the pump-shell, which increases the clearance between the shell and the wing, quickly impairing its efficiency.

In this piston the sand is taken in at the centre and is deposited in the pump-shell, which has a clearance at the periphery of the same equal to the diameter of the pump's discharge. When the pump is in operation this is stated by the makers to form a water-cushion equal at all points on the outer edge of the piston, which is not found in any other pump.

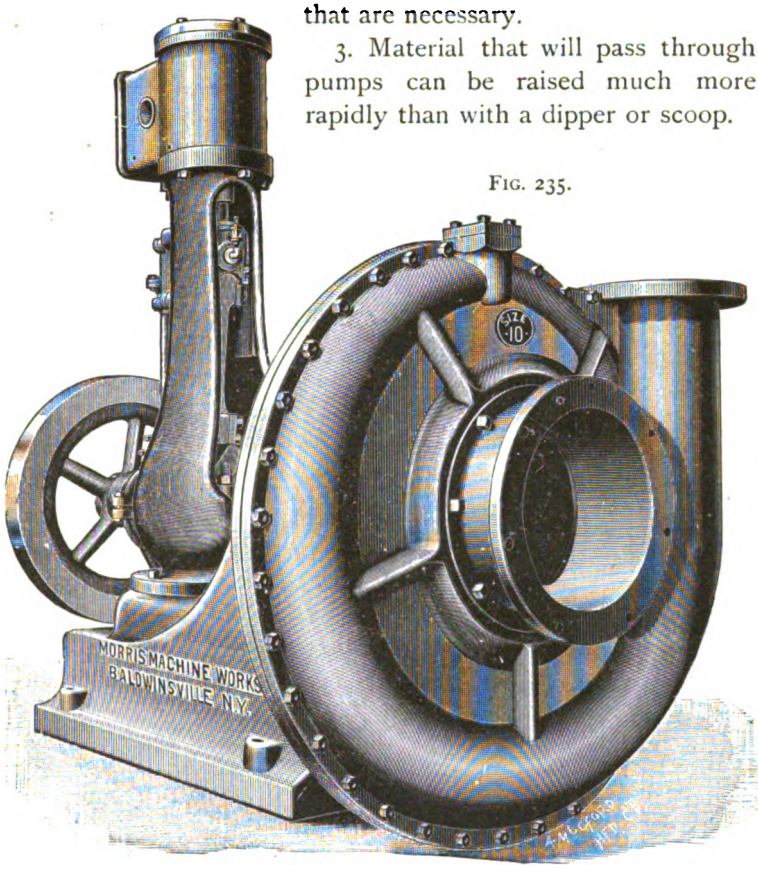
It is said that sand, mud, gravel, or marl can be dredged with a centrifugal pump at a much less cost than with the old-style dipper- or scoop-dredge, for the following reasons:

1. The plant costs about one-twentieth as much as a dipper-dredge.

2. Less help required to run it, two or three men being all that are necessary.

3. Material that will pass through pumps can be raised much more rapidly than with a dipper or scoop.

FIG. 235.



4. That a pump-dredge can work in weather and sea where a dipper-dredge could not live.

5. That a centrifugal plant is not liable to the many breakages that occur in the dipper machines.

6. When working in rivers or bays, material can be dredged and piped ashore at one operation.

TABLE XXVII.

SPEED AT WHICH MORRIS MACHINE-WORKS CENTRIFUGAL PUMPS SHOULD RUN TO RAISE WATER TO DIFFERENT HEIGHTS.

Number.	Feet. 5.	Feet. 10.	Feet. 15.	Feet. 20.	Feet. 25.	Feet. 30.	Feet. 35.	Feet. 40.	Feet. 50.	Feet. 60.	Feet. 70.	Feet. 80.	Feet. 90.	Feet. 100.
1½	426	603	740	850	955	1046	1130	1211	1350	1478	1597	1705	1812	1905
1¾	341	484	592	683	765	837	905	977	1081	1216	1282	1368	1451	1530
2	323	460	564	650	727	796	886	920	1028	1127	1217	1302	1381	1456
2½	314	446	546	630	705	772	835	892	997	1092	1180	1262	1337	1411
3	300	426	521	602	660	737	796	852	952	1043	1127	1205	1277	1322
4	283	398	480	557	621	693	752	808	895	992	1054	1138	1220	1275
5	250	357	437	505	565	619	692	715	800	876	946	1012	1073	1128
6	215	300	370	426	485	522	562	609	675	745	790	860	940	965
8	182	258	316	365	408	447	483	518	577	633	686	730	776	835
10	168	237	291	336	375	411	443	475	531	582	626	671	712	751
12	133	189	231	267	300	327	353	378	423	463	501	536	568	600
15	108	154	189	216	242	266	287	307	343	377	407	436	462	487
18	92	135	160	180	203	231	246	277	304	323	352	374	400	413
22	80	112	139	160	178	197	212	227	253	277	300	321	340	358

The application of a centrifugal pump to dredging operations was tested by General Q. A. Gillmore, U. S. A., in deepening the channel over the bar at the mouth of the St. John's River, Florida. A No. 9 Andrews centrifugal drainage-pump was used on the deck of a small steamer. This pump had suction- and discharge-openings 9 inches in diameter.

To the suction-opening there are connected by a 2-way branch-pipe two 6-inch suction-pipes, instead of one 9-inch as usual, the object being not only to work on both sides of the boat simultaneously, but to render the necessary handling of the pipes as easy and prompt as possible. There is, on the other hand, considerable disadvantage in operating with two suction-pipes instead of one, on account of the greater amount of friction for an equivalent suction-capacity; for while a 9-inch pipe has an area of 81 circular inches, two 6-inch pipes have an aggregate area of only 72 circular inches. The frictional surface is therefore increased as 27 to 36, making the disadvantage from or loss by friction from this cause as 2 to 3.

As a partial compensation for this increased amount of friction, an increased velocity is given to water in suction-pipes of less aggregate area than the discharge-pipe, and a larger proportion of sand is thereby carried up.

It was necessary also to encounter another disadvantage by using several bends, of which there were two in each of the suction-pipes, and one in the discharge-pipe, those in the suction-pipe being each one-eighth of a circle, and that in the discharge-pipe one-fourth of a circle. These bends reduce the delivery at the rate of 10 per cent. for each turn of 90 degrees, and about 6 per cent. for each turn of 45 degrees, the reductions in each case being calculated upon the quantity passing the preceding bend.

Thus, the first one-eighth bend in the suction reduces the quantity to 94 per cent., the second to 88 per cent., and the one-fourth bend in the discharge to 79 per cent. The disadvantages, therefore, under which the apparatus labored may be briefly summed up as follows:

1. The loss by friction, due to the use of two 6-inch instead of one 9-inch suction-pipe, is increased 50 per cent.

2. The unestimated loss by friction, due to the use of suction-pipes three times as long as the height to which the material is to be raised.

3. The loss of 21 per cent. by bends in the suction- and discharge-pipes.

Although 200 revolutions of the pump-disk per minute will easily raise 3000 gallons of clear water 12 feet high through a straight vertical 9-inch pipe, 300 revolutions are required to raise 2500 gallons of sand and water 11 feet high, through the two inclined suction-pipes having two turns each, discharged through a pipe having one turn. To prevent the ends of the suction-pipes being lifted off the bottom by the pitching of the boat, and as a precaution against accident, a portion of each pipe is made flexible, being composed of 6-inch rubber hose stretched over a coil of wire. In addition, the ends are loaded with an iron frame or drag, each weighing about 250 pounds, which is intended to move flat along the bottom during the operation of dredging. To the under surface of this frame, directly below the mouth of the pipe, a number of teeth or knives are attached to stir up the sand and aid its entrance into the pipes. A chain attached to each drag, and leading to the deck of the steamer on either side, takes the strain from the pipe when the drag is down and the steamer in motion.

The proportion of sand that can be pumped depends greatly upon its specific gravity and fineness. The calcareous and argillaceous sands flow more freely than the silicious, and fine sands are less liable to choke the pipe than those that are coarse. When working at high speed, 50 to 55 per cent. of sand can easily be raised through a straight vertical pipe, giving for every 10 cubic yards of material discharged 5 to 5½ cubic yards of compact sand. With the appliances used on the St. John's bar the proportion of sand seldom exceeded 45 per cent., generally ranging from 30 to

35 per cent. when working under the most favorable conditions.

In pumping 2500 gallons, or 12.6 cubic yards, of sand and water per minute, we would therefore get from 3.7 to 4.3 cubic yards of sand. During the early stages of the work, before the teeth under the drag had been properly arranged to aid the flow of sand into the pipes, the yield was considerably below this average, not often exceeding, and frequently falling below, 2 cubic yards of sand per minute during the time actually occupied in pumping.

With a centrifugal drainage-pump sand can be easily discharged at a height of 30 feet above the level of the water; and when the distance to which it has to be conveyed is so great that open troughs from the discharge-pipe to the dumping-ground cannot have sufficient inclination to secure a free flow of the sand and water, it would be necessary to make the discharge through pipes, increasing the power expended in proportion to their length, so as to insure a velocity that will transport the sand and prevent choking. The pump itself should, in all cases, be placed as low as possible, and it would generally be practicable to locate it from 3 to 5 feet above the surface of the water.

The cost of dredging with a 9-inch pump would probably not exceed 10 to 11 cents per cubic yard, inclusive of running expenses, wear and tear of machinery, and all stoppages for repairs and other contingencies. Indeed, assuming the pump on St. John's Bar to have worked continuously in raising sand 10 hours per day, except Sundays, with the same average results per hour actually attained while pumping, thus charging the six working-days of each week with the expense of seven, the cost of raising the sand into the bins would have been only $8\frac{3}{10}$ cents per cubic yard, and if it could, at the same time, have also been continuously discharged to the dumping-ground, through either open troughs or pipes, no additional expense, except a trifle for increased power, would have been incurred. There were, moreover, constant losses

encountered on the bar while actually pumping, which would not occur in still water, and of which no account has been taken, due to the pitching of the boat, which frequently lifted the ends of the suction-pipes from the bottom. It is therefore considered safe to estimate the cost of removing sand at 10 to 11 cents per cubic yard, when the conditions are such that the work of raising the sand and discharging to the dumping-ground can be carried on simultaneously and continuously.

Centrifugal Pumps in Series.—Such a method of pumping water is not often carried out, but a recent example of such an application was shown in an instance where it was necessary to raise water 150 feet, the only appliances available for this purpose being a pair of centrifugal pumps, either of which was not equal to raising the water to a height above 75 feet. It should be remembered that the operation of a centrifugal pump is due entirely to the centrifugal force of the water as it is revolving in the pump-case, but the water was raised to the required height by attaching the two pumps in series, the second one adding the necessary increment to the pressure which was received, and working in that way very satisfactorily.

The reader is referred to an interesting and valuable paper on the "Irrigating Machinery of the Pacific Coast," by Mr. John Richards, published in the Proceedings of the Institution of Mechanical Engineers, London, 1888. This paper was also published in *Engineering*, Vol. XLIV. The paper is fully illustrated, and the writer only regrets that want of space prevents its reproduction here, as Mr. Richards has kindly placed it at his disposal.

CHAPTER XVII.

DUTY-TRIALS OF PUMPING ENGINES.

THE duty of a pumping engine is a term used to denote the number of pounds of water lifted 1 foot high by the combustion of 100 pounds of coal. This is a convenient method by which to compare the results of pumping engines one with another, because a measure of duty in so many millions of foot-pounds of work done fixes at once the relative position which such an engine occupies when compared with other pumping engines.

The method of calculation usually employed in ascertaining the duty of a pumping engine may be expressed in the following formula:

$$\frac{A \times P \times V \times 100}{C} = \text{Duty.}$$

Where A = area of plunger, in square inches.
P = pressure in pounds per square inch.
V = total plunger-travel during the trial, in feet.
C = coal burnt during the trial, in pounds.

Example based on a 10 hours' trial:

A = 201 square inches, area of plunger.
P = 80 pounds pressure, including the suction-lift.
V = 66,000 feet of plunger-travel in 10 hours.
C = 1600 pounds of coal burnt in 10 hours.

Then $\frac{201 \times 80 \times 66{,}000 \times 100}{1600} = 66{,}330{,}000$ foot-pounds of duty.

Another method of determining the duty of the same pumping engine may be expressed thus:

$$\frac{G \times W \times H \times 100}{C} = \text{Duty.}$$

Where G = gallons of water actually delivered.
W = weight of water per gallon.
H = height in feet to which the water is pumped.
C = coal burnt during the trial, in pounds.

Example based on a 10 hours' trial:

$G = 689{,}040$ gallons in 10 hours.
$W = 8.33$ pounds per gallon.
$H = 184$ feet height to which the water is pumped.
$C = 1600$ pounds of coal burnt in 10 hours.

Then $\frac{689{,}040 \times 8.33 \times 184 \times 100}{1600} = 66{,}006{,}587$ foot-pounds of duty.

A high-duty pumping engine is understood to be one in which nearly or more than 100,000,000 foot-pounds of work are accomplished with an expenditure of 100 pounds of coal burnt in the furnace. This definition, it will be seen, couples the boiler and pumping engine as one machine, the coal being charged against one end of it, and the foot-pounds of work done are credited at the other; and so far as the average water-works management goes, it is about all they care to know. To the engineer there are, however, intermediate problems which are not only interesting in themselves, but useful to know, that a correct estimate may be reached regarding the efficiency of the plant in its several details.

The boiler part of the pumping plant should always be tested separately from that of the pumping engine. In duty-trials it has been customary to assume the evaporation of 10 pounds of water from and at 212° Fahr. per pound of coal, without any allowance for ashes and clinkers. This assumption is not always realized in practice, and a deduction of from 10 to 20 per cent. must be made to have it accord with ordinary boiler evaporation. This fictitious standard of boiler performance probably originated in a divided responsibility, one contractor furnishing the boilers and another the pumping engine. Some contracts have been made to read 1000 pounds of dry steam delivered to the engines, rather than the more usual one of 100 pounds of coal coupled with a boiler performance of 10 pounds of water evaporated per pound of coal.

The boiler-test should be conducted independently of the engine, because a low efficiency is more likely to occur there than in the engine; the ordinary performance of the latter being such that calculations may be made in advance with close ap-

proximation to experimental results. There are many things which affect boiler performance: the boiler may not be properly proportioned, the circulation in it not good, the furnace may not be the best for the fuel to be used, the draft may be deficient on the one hand, and too much air may be admitted for the weight of fuel burnt on the other. The heating-power of coal must of necessity be variable, because no two mines yield exactly the same quality of coal, to say nothing of such wide divergencies as exist between the properties of anthracite and bituminous coals; it is clear, therefore, that the coal standard is at best a very uncertain one in estimating the efficiency of a pumping-plant.

The Duty-Trial Committee in their report to the American Society of Mechanical Engineers, published in Vol. XI., Transactions, recommend that in all reports on duty-trials of pumping engines, the existing unit of 100 pounds of coal be abolished, and that in its place a new basis of 1,000,000 heat-units be established. This new unit will not seriously affect comparison with the one abolished, because it is expected that 1 pound of coal will yield at least 10,000 heat-units; therefore $\frac{10{,}000}{966} = 10.35$ pounds of water evaporated from and at 212° Fahr., the denominator 966 being the latent heat of evaporation.

The final results of a boiler-trial, for the sake of uniformity and comparison with other trials should, as far as practicable, be tabulated in the form recommended by the American Society of Mechanical Engineers, Transactions, Vol. XI., which is here reproduced.

1. Date of trial . . . . . .
2. Duration of trial . . . . . Hours.

DIMENSIONS AND PROPORTIONS.

3. Grate surface wide long Area . . . . . Square feet.
4. Water-heating surface . . . . . "
5. Superheating surface . . . . . "
6. Ratio of water-heating surface to grate surface . . . . . .

(Give also complete description of boilers.)

AVERAGE PRESSURES.	
7. Steam-pressure in boiler by gauge	Pounds.
8. Atmospheric pressure by barometer	"
9. Force of draught in inches of water	Inches.
AVERAGE TEMPERATURES.	
10. Of steam	Degrees.
11. Of escaping gases	"
12. Of feed water	
FUEL.	
13. Total amount of coal consumed*	Pounds.
14. Moisture in coal	Per cent.
15. Dry coal consumed	Pounds.
16. Total refuse (dry)	"
17. Total combustible (dry weight of coal, item 15, less refuse, item 16)	"
18. Dry coal consumed per hour	"
RESULTS OF CALORIMETRIC TEST.	
19 Quality of steam, dry steam being taken as unity	
20. Percentage of moisture in steam	Per cent.
21. Number of degrees superheated	Degrees.
WATER.	
22. Total weight of water pumped into boiler and apparently evaporated†	Pounds.
23. Water actually evaporated corrected for quality of steam	"
24. Equivalent water evaporated into dry steam from and at 212° Fahr.‡	"
25. Equivalent total heat derived from fuel, in British Thermal Units	B.T.U.
26. Equivalent water evaporated into dry steam from and at 212° Fahr. per hour	Pounds.
ECONOMIC EVAPORATION.	
27. Water actually evaporated per pound of dry coal from actual pressure and temperature	Pounds.
28. Equivalent water evaporated per pound of dry coal from and at 212° Fahr.	"

* Including equivalent of wood used in lighting fire. One pound of wood equals 0.4 of a pound of coal, not including unburned coal withdrawn from fire at end of test.

† Corrected for inequality of water-level and of steam-pressure at beginning and end of test.

‡ Factor of evaporation $= \frac{H-h}{965.7}$, H and h being respectively the total heat-units in steam of the average observed pressure, and in water of the average observed temperature of feed.

29. Equivalent water evaporated per pound of combustible from and at 212° Fahr. . . . . Pounds.
30. Number of pounds of coal required to supply 1,000,000 British Thermal Units . . . . . "

RATE OF COMBUSTION.

31. Dry coal actually burned per square foot of grate surface per hour Pounds.

RATE OF EVAPORATION.

32. Water evaporated from and at 212° Fahr. per square foot of heating surface per hour . . . . . Pounds.

[*Note.*—To determine the percentage of surface-moisture in the coal a sample of the coal should be dried for a period of twenty-four hours, being subjected to a temperature of not more than 212°. The quantity of unconsumed coal contained in the refuse withdrawn from the furnace and ash-pit at the end of the test may be found by sifting either the whole of the refuse or a sample of the same, in a screen having ⅛-inch meshes. This, deducted from the weight of dry coal fired, gives the weight of dry coal consumed, for line 15.—*Duty-Trial Committee.*]

The pumping engine duty-trial should be conducted on the lines proposed by the American Society of Mechanical Engineers, previously referred to.

The heat-unit basis of computing duty is proposed by the above society, and to make the computation from the quantity of heat supplied to the complete plant; using not only that supplied to the engine-cylinders, but that supplied to all the accessory parts of the engine, such as the steam-jackets, the donkey feed-pump, the independent air-pump, if this be driven with steam, and any other apparatus using steam which is necessary to the operation of the engine. It is recommended that the scope of the test be made so broad that, for the sake of completeness, the quantity of steam which passes through the cylinders of the engine be determined independently of that used for other purposes, and likewise, that the quantity of steam used by each accessory part of the engine be also determined. In contract tests, if a steam-pump be used for the boiler feed-pump, the quantity of heat supplied for operating this apparatus is to be included in the

total quantity, not only in cases where both boiler and engine are supplied by one party, but also where the boiler is furnished by a separate contractor.

The heat-unit method requires that the actual total heat of the steam shall be known, and for this purpose allowance will necessarily be made for any moisture or superheat contained by the steam furnished to the engine.

Measuring the Work done.—In determining upon a suitable method of measuring the amount of work done, which involves a measure of the quantity of water discharged into the force main, the committee have endeavored to find one which may be employed universally, and which may, in a reasonable manner, serve the ends of the builder, purchaser, and all interested parties.

It is proposed that the plunger-displacement system of measurement be employed, and that the purchaser's interest be protected by the determination of the amount of slip in the pump, so far as slip is produced by leakage of the plunger, which is probably its main factor, and leakage of valves, if this occurs through faulty design. A satisfactory determination of the approximate extent to which leakage occurs does not present serious difficulty.

In deciding upon plunger displacement, accompanied by a determination of the leakage, as the best mode of measurement for the purposes in view, the committee does not for a moment underrate the importance and desirability of measurement by weir, tube, or nozzle, whenever either of these can be employed to advantage. It is strongly recommended that these additional measurements be undertaken in all cases where it is practicable to do so, that the results of the test may be supplemented by the additional data thus obtained.

Quantity of Work done.—In determining the quantity of work done by the pump, the committee recommends that the work of overcoming the friction of the water in passing through the passages and valves in the pump should not be

included in the desired total; but that the work expended in friction of both the force- and suction-mains be included in that on which the duty is computed. It is held that the efficiency of the engine should not be made dependent upon any condition which is foreign to itself, and that the builder of the engine should be held responsible only for the work done from the time when the water enters the pump to the time when it leaves it. The purchaser, it should be observed, should guard his interest in the matter by having the mains furnished of such capacity as to reduce their friction to a minimum.

To carry out these provisions, the data to be determined, apart from that relating to the plunger displacement, are the indication of a pressure-gauge attached to the force-main, that of a vacuum-gauge attached to the suction-main, and the vertical distance between the centres of the two gauges.

It is recommended that no air be allowed to enter the pump-cylinders during the progress of the test, thereby removing all possibility of imperfect filling. If it is necessary, in special cases, for air to be "snifted in," this should be regarded as a defect in the action of the pump, which should be noted by the expert in his report, and such allowance should be made for the imperfect filling, due to the presence of air, as may be determined upon by an examination of the indicator diagrams taken from the pump-cylinders, or from other data which may be secured.

Engine Performance.—The necessary data having been obtained in accordance with these recommendations, the duty of an engine, and other quantities relating to its performance, may be computed by the use of the following formulæ:

1. $$\text{Duty} = \frac{\text{Foot-pounds of work done}}{\text{Total number of heat-units consumed}} \times 1{,}000{,}000$$

$$= \frac{A\,(P \pm p + s) \times L \times N}{H} \times 1{,}000{,}000 \text{ (foot-pounds).}$$

2. $$\text{Percentage of leakage} = \frac{C \times 144}{A \times L \times N} \times 100 \text{ (per cent).}$$

3. Capacity = number of gallons of water discharged in 24 hours

$$= \frac{A \times L \times N \times 7.4805 \times 24}{D \times 144}$$

$$= \frac{A \times L \times N \times 1.24675}{D} \text{ (gallons).}$$

4. Percentage of total frictions

$$= \left[\frac{I.H.P. - \frac{A\,(P \pm p + s) \times L \times N}{D \times 60 \times 33{,}000}}{I.H.P.} \right] \times 100$$

$$= \left[1 - \frac{A\,(P \pm p + s) \times L \times N}{As \times M.E.P. \times Ls \times Ns} \right] \times 100 \text{ (per cent.);}$$

or, in the usual case, where the length of the stroke and number of strokes of the plunger are the same as that of the steam piston, this last formula becomes:

$$\text{Percentage of total frictions} = \left[1 - \frac{A\,(P \pm p + s)}{As \times M.E.P.} \right] \times 100 \text{ (per cent.).}$$

In these formulæ the letters refer to the following quantities:

A = Area, in square inches, of pump-plunger or piston, corrected for area of piston-rod. (When one rod is used at one end only the correction is one-half the area of the rod. If there is more than one rod the correction is multiplied accordingly.)

P = Pressure, in pounds per square inch, indicated by the gauge on the force-main.

p = Pressure, in pounds per square inch, corresponding to indication of the vacuum-gauge on suction-main (or pressure-gauge if the suction-pipe is under a head). The indication of the vacuum-gauge, in inches of mercury, may be converted into pounds by dividing it by 2.035.

s = Pressure, in pounds per square inch, corresponding to distance between the centres of the two gauges. The computation for this pressure is made by multiplying the distance, expressed in feet, by the weight of one cubic foot of water at the temperature of the pump-well, and dividing the product by 144, or by multiplying the distance in feet by the appropriate quantity found in the following table. The quantities in this table are computed from the weights of one cubic foot of water at the various temperatures, as given by D. K. Clark in his "Rules and Tables," which also correspond to Charles T. Porter's figures in his work on the "Richards's Steam-Engine Indicator."

Temperature of Water in Pump-Well.	Weight of 1 Cubic Foot of Water divided by 144.	Temperature of Water in Pump-Well.	Weight of 1 Cubic Foot of Water divided by 144.
Degrees Fahrenheit.		Degrees Fahrenheit.	
32	.4335	75	.4325
35	.4335	80	.4322
40	.4335	85	.4319
45	.4334	90	.4315
50	.4333	95	.4311
55	.4332	100	.4307
60	.4331	105	.4303
65	.4329	110	.4298
70	.4327		

L = Average length of stroke of pump-plunger in feet.

N = Total number of single strokes of pump-plunger made during the trial.

As = Area of steam-cylinder, in square inches, corrected for area of piston-rod. The quantity, $As \times M.E.P.$, in an engine having more than one cylinder, is the sum of the various quantities relating to the respective cylinders.

Ls = Average length of stroke of steam-piston, in feet.

Ns = Total number of single strokes of steam-piston during trial.

$M.E.P.$ = Average mean effective pressure in pounds per square inch, measured from the indicator diagrams taken from the steam-cylinder.

$I.H.P.$ = Indicated horse-power developed by the steam-cylinder.

C = Total number of cubic feet of water which leaked by the pump-plunger during the trial, estimated from the results of the leakage test.

D = Duration of trial, in hours.

H = Total number of heat-units [*B. T. U.*] consumed by engine = weight of water supplied to boiler by main feed-pump × total heat of steam of boiler-pressure reckoned from temperature of main feed-water + weight of water supplied by jacket-pump × total heat of steam of boiler-pressure reckoned from temperature of jacket-water + weight of any other water supplied × total heat of steam reckoned from its temperature of supply. The total heat of the steam is corrected for the moisture or superheat which the steam may contain. For moisture, the correction is subtracted, and is found by multiplying the latent heat of the steam by the percentage of moisture, and dividing the product by 100. For superheat, the correction is added, and is found by multiplying the number of degrees of superheating (*i.e.*, the excess of the temperature of the steam above the normal temperature of saturated steam) by 0.48. No allowance is made for heat added to the feed-water, which is derived from any source, except the engine or some accessory of the engine. Heat added to the water by the use of a flue heater at the boiler is not to be deducted. Should heat be abstracted from the flue by means of a steam reheater connected with the intermediate receiver of the engine, this heat must be included in the total quantity supplied by the boiler.

The total and latent heats may be found by reference to the Tables of the Properties of Saturated Steam, given in Charles T. Porter's treatise on the "Richards's Steam-Engine Indicator."

The two examples following, (1) compound direct-acting duplex pumping engine, and (2) compound crank and flywheel pumping engine, are given to illustrate the method of computation. The figures are not obtained from tests actually made, but they correspond in round numbers with those which were so obtained.

Compound Tandem Direct-Acting Duplex Pumping Engine.—Both high-pressure and low-pressure cylinders jacketed with live steam. Jet condenser used, with air-pump driven by main engine. Boiler feed-pump also driven by main engine. Jacket-water returned to boiler by gravity. Main supply of feed-water drawn from hot well.

DIMENSIONS.

Diameter of each high-pressure cylinder (two)	15 inches.
Diameter of each low-pressure cylinder (two)	30 "
Diameter of piston-rod, each cylinder (one at each end high-pressure, two at one end low-pressure)	3.5 "
Diameter of pump-plungers (two)	15 "
Diameter of piston-rod, each plunger (one at one end)	3.5 "
Nominal stroke	18 "

GENERAL DATA.

1. Duration of test (D)	12 hours.
2. Boiler-pressure by gauge (barometric-pressure, 14.7 pounds)	120 pounds.
3. Temperature of water in pump-well	80 deg.
4. Temperature of main supply of feed-water	100 "
5. Temperature of jacket-water	280 "
6. Percentage of moisture in steam	0 per ct.
7. Weight of water supplied to boiler by main feed-pump	22,400 pounds.
8. Weight of water supplied to boiler by jackets	2560 "

DATA RELATING TO WORK OF PUMP.

9. Area of plunger minus ½ area of rod (A)	171.9 sq. in.
10. Average length of stroke (L and Ls)	1.572 feet.
11. Total number of single strokes during trial (N and Ns)	76,000
12. Pressure by gauge on force-main (P)	100 pounds.

13. Vacuum by gauge on suction-main	9.3 inches.
14. Pressure corresponding to vacuum given in preceding line (p)	4.57 pounds.
15. Vertical distance between gauges	8 feet.
16. Pressure corresponding to distance given in preceding line (s)	3.46 pounds.
17. Volume of water which leaked through the plungers computed from results of leakage test (C)	5900 cu. ft.

DATA RELATING TO WORK OF STEAM-CYLINDERS.

18. Area of high-pressure piston minus area of one rod (As_1) .	167.09 sq. in.
19. Mean effective pressure high-pressure cylinder ($M.E.P._1$)	61.31 pounds.
20. Area of low-pressure piston minus ½ area of two rods (As_2)	697.24 sq. in.
21. Mean effective pressure low-pressure cylinder ($M.E.P._2$)	13.72 pounds.
22. Number of double strokes each side per minute	26.39
23. Indicated horse-power developed by steam-cylinders . .	99.61 I.H.P.
24. Feed-water consumed per indicated horse-power per hour	20.88 pounds.
25. Heat-units consumed per indicated horse-power per hour . . . . . = 383 B.T.U. per minute.	23,088 B.T.U.

TOTAL HEAT OF STEAM RECKONED FROM THE VARIOUS TEMPERATURES OF FEED-WATER AND COMPUTATIONS BASED THEREON.

26. Total heat of 1 pound of dry steam at 120 pounds gauge-pressure reckoned from 0° Fahr.	1220.6 B.T.U.
27. Ditto, reckoned from temperature of main feed-water (100°)	1120.5 "
28. Ditto, reckoned from temperature of jacket-water (280°)	938.5 "
29. Heat consumed by engine (H) (22,400 × 1120.5) + (2560 × 938.5)	27,501,760

RESULTS.

Substituting these quantities in the formulæ, we have

$$1.\ \text{Duty} = \frac{171.9 \times (100 + 4.57 + 3.46) \times 1.572 \times 76{,}000}{27{,}501{,}760} \times 1{,}000{,}000$$

$$= 80{,}671{,}622 \text{ foot-pounds.}$$

$$2.\ \text{Percentage of leakage} = \frac{5{,}900 \times 144}{171.9 \times 1.572 \times 76{,}000} \times 100 = 4.1 \text{ per cent.}$$

3. Capacity $= \dfrac{171.9 \times 1.572 \times 76{,}000 \times 1.24675}{\frac{D}{12}} = 2{,}133{,}735$ gallons.

4. Percentage of total frictions

$$= \left[1 - \frac{171.9 \times (100 + 4.57 + 3.46)}{(167.09 \times 61.31) + (697.24 \times 13.72)} \right] \times 100 = 9.4 \text{ per cent.}$$

Compound Fly-Wheel Pumping Engine.—High-pressure cylinder jacketed with live steam from the boiler. Low-pressure cylinder jacketed with steam from the intermediate receiver, the condensed water from which is returned to the boiler by means of a pump operated by the engine. Main steam-pipe fitted with a separator. The intermediate receiver provided with a reheater supplied with boiler steam. Water drained from high-pressure jacket, separator, and reheater, collected in a closed tank under boiler-pressure, and from this point fed to the boiler direct by an independent steam-pump. Jet condenser used operated by an independent air-pump. Main supply of feed-water drawn from hot well and fed to the boiler by donkey steam-pump, which discharges through a feed-water heater. All the steam-pumps, together with the independent air-pump, exhaust through the heater to the atmosphere.

DIMENSIONS.

Diameter of high-pressure steam-cylinder (one)	20 inches.
Diameter of low-pressure steam-cylinder (one)	40 "
Diameter of plunger (one)	20 "
Diameter of each piston-rod	4 "
Stroke of steam-pistons and pump-plunger	3 feet.

GENERAL DATA.

1. Duration of trial (D)	10 hours.
2. Boiler-pressure indicated by gauge (barometric pressure, 14.7 pounds)	120 pounds.
3. Temperature of water in pump-well	60 deg.
4. Temperature of water supplied to boiler by main feed-pump, leaving heater	215 "
5. Temperature of water supplied by low-pressure jacket-pump	225 "

6.	Temperature of water supplied by high-pressure jacket, separator, and reheater pump, that derived from separator being 340°, and that from jackets 290°	300 deg.
7.	Weight of water supplied to boiler by main feed-pump .	18,863 pounds.
8.	Weight of water supplied by low-pressure jacket-pump .	615 "
9.	Weight of water supplied by pump for high-pressure jacket, separator, and reheater tank, of which 210 pounds is derived from separator	1025 "
10.	Total weight of feed-water supplied from all sources . .	20,503 "
11.	Percentage of moisture in steam after leaving separator .	1.5 per ct.

DATA RELATING TO WORK OF PUMP.

12.	Area of plunger minus ½ area of piston-rod (*A*) . . .	307.88 sq. in.
13.	Average length of stroke (*L* and *Ls*)	3 feet.
14.	Total number of single strokes during trial (*N* and *Ns*) .	24,000
15.	Pressure by gauge on force-main (*P*)	95 pounds.
16.	Vacuum by gauge on suction-main	7.5 inches.
17.	Pressure corresponding to vacuum given in preceding line (*p*)	3.69 pounds.
18.	Vertical distance between centres of two gauges	10 feet.
19.	Pressure equivalent to distance between two gauges (*s*) .	4.33 pounds.
20.	Total leakage of pump during trial, determined from results of leakage test (*C*)	3078 cu. feet.
21.	Number of double strokes of pump per minute	20
22.	Mean effective pressure measured from pump diagrams .	105 pounds.
23.	Indicated horse-power exerted in pump cylinders . . .	117.55 I.H.P.

DATA RELATING TO WORK OF STEAM-CYLINDERS.

24.	Area of high-pressure piston minus ½ area of rod (As_1) .	307.88 sq. ins.
25.	Area of low-pressure piston minus ½ area of rod (As_2) .	1250.36 "
26.	Average length of stroke, each	3 feet.
27.	Mean effective pressure measured from high-pressure diagrams ($M.E.P._1$)	59.25 pounds.
28.	Mean effective pressure measured from low-pressure diagrams ($M.E.P._2$)	13.60 "
29.	Number of double strokes per minute (line 21)	20
30.	Indicated horse-power developed by high-pressure cylinder	66.33 I.H.P.
31.	Indicated horse-power developed by low-pressure cylinder	61.82 "
32.	Indicated horse-power developed by both cylinders . .	128.15 "
33.	Feed-water consumed by plant per indicated horse-power per hour, corrected for separator water and for moisture in steam	15.60 pounds.
34.	Number of heat-units consumed per indicated horse-power per hour	15,652.1 B.T.U.
35.	Number of heat-units consumed per indicated horse-power per minute	260.9 "

TOTAL HEAT OF STEAM RECKONED FROM THE VARIOUS TEMPERATURES OF FEED-WATER AND COMPUTATIONS BASED THEREON.

36. Total heat of 1 pound of steam at 120 pounds gauge-pressure, containing 1.5 per cent. of moisture, reckoned from 0° Fahr. = 1220.6 — (1.5 per cent. of 866.7) .	1207.6	B.T.U.
37. Ditto, reckoned from 215°, temperature of main feed-water = 1207.6 — 215.9	991.7	"
38. Ditto, reckoned from 225°, temperature of low-pressure jacket-water = 1207.6 — 226.1	981.5	"
39. Ditto, reckoned from 290°, temperature of high-pressure jacket and reheater water = 1207.6 — 292.3	915 3	"
40. Heat of separator water reckoned from 340° = 353.9 — 343.8	10.1	"
41. Heat consumed by engine (H) = (18,863 × 991.7) + (615 × 981.5) + (815 × 915.3) + (210 × 10.1) . .	20,058,150	"

RESULTS.

Substituting these quantities in the formulæ, we have:

1. Duty $= \dfrac{\overset{A}{307.88} \times (95 + \overset{P}{3.69} + \overset{p}{4.33}) \times \overset{s}{3} \times \overset{L}{24,000}}{\underset{H}{20,058,150}} \times 1,000,000$

$= 113,853,044$ foot-pounds.

2. Percentage of leakage $= \dfrac{\overset{C}{3078} \times 144}{\underset{A}{307.88} \times \underset{L}{3} \times \underset{N}{24,000}} \times 100 = 2$ per cent.

3. Capacity $= \dfrac{\overset{A}{307.88} \times \overset{L}{3} \times \overset{N}{24,000} \times 1.24675}{\underset{10}{D}} = 2,763,716$ gallons.

4. Percentage of total frictions

$$= \left[1 - \frac{\overset{A}{307.88} \times (95 + \overset{P}{3.69} + \overset{p}{4.33}) \times \overset{s}{3}}{\underset{As_1}{(307.88 \times 59.25)} + \underset{M.E.P._1}{(1250.36 \times 13.6)} \quad \underset{As_2}{\quad} \quad \underset{M.E.P._2}{\quad}} \right] \times 100 = 9 \text{ per ct.}$$

Guaranteed Performance.—In order that the contract between builder and purchaser of a pumping engine may conform to the proposed standard, the guarantee as to performance should be expressed in the following terms:

1. The engine shall perform a duty, based upon plunger displacement, equivalent to not less than . . . foot-pounds of work for each one million heat-units consumed.

2. The leakage of the pump shall not exceed . . . per cent. of the total plunger displacement when the engine is working at its rated capacity.

3. The boiler shall supply one million heat-units to the engine on a consumption of . . . pounds of . . . coal, or it shall evaporate not less than . . . pounds of water from and at 212 degrees per pound of the combustible portion of the coal named.

4. The mode of determining these quantities is to conform to the standard method of conducting duty-trials recommended by the Committee of the American Society of Mechanical Engineers.

Should one contractor furnish the engine and another the boiler, separate guarantees will be made, the individual requirements of which are the same as those noted.

It is desirable, where both parties concur therein, to introduce into the contract the following additional provision regarding friction,—viz.:

"The friction of the engine shall not exceed . . . per cent. of the indicated power developed in the steam-cylinders."

The general mode of operation is to first subject the plant to a preliminary run under the working conditions, for the purpose of determining the temperature of the feed-water or the several temperatures where there is more than one supply. It is usually impracticable to weigh the main supply of water, derived, as it generally is, from a low-placed hot well, and the test of the main quantity of feed-water used must, as a rule, be made with cold water drawn from the service-main. The changed conditions in the working of the plant thus introduced and the arrangement of apparatus which is frequently needed to measure the additional supplies of feed-water, make it desirable to obtain the working temperatures as a preliminary to the main duty-trial. Hence the preliminary run is made, as noted, merely for securing the temperatures. The main test of the boiler and engine is then carried forward, and during this test the weights of the various supplies of feed-water are determined, and the remaining data needed for making the computations. Finally, as soon as practicable after these tests are completed, the rate of leakage through the pump is measured with the engine at rest.

As to the duration of the test, it appears to the committee

that, so far as the main trial is concerned, which is practically a feed-water test, it need not be prolonged more than ten hours, unless, in that time, appreciable errors should be produced by inaccuracies in the observations of the height of water in the gauge-glass. The duration of the boiler-trial might, with good reason, be made longer were it not that the results of the boiler-test are independent of those of the duty-trial. It is desirable to reduce, if possible, the number of hours of the trial to such a point that the time expended upon the work, including that required in preparation for the beginning of the test and that spent in bringing the test to a close, shall be such that the same expert, without undue physical exertion, may have the test under his continuous supervision from beginning to end. This is feasible where the length of the duty-trial, according to the plan proposed, does not exceed ten hours.

An English method of determining the duty of a pumping engine is shown in the annexed abstract of a report on a Worthington high-duty pumping engine. The test was conducted by Professor W. C. Unwin; the engine was constructed by Messrs. James Simpson & Co., London. The report is interesting for two reasons,—first, as showing the economy of a large pumping engine on a low lift, using a steam-pressure much lower than is common with us in high-duty engines; and, second, as showing Professor Unwin's method of conducting a duty-trial, which differs in some respects from the methods which obtain in this country.

The engines are of the compound Worthington duplex type, fitted with high-duty attachment, the main dimensions being as follows:

Diameter high-pressure pistons	27 inches.
" low-pressure pistons	54 "
" pump-plungers	40 "

All of 44 inches stroke.

The valves are of india-rubber and spring-loaded. The compensating cylinders are 11 inches diameter, and loaded by air-pressure to about 120 pounds per square inch. The pumps lift water from a well communicating with the river and deliver it through two 3-foot mains to the reservoir about nine miles distant. The head during these trials, measured by the difference of pressure in the suction- and discharge-pipes, was from 50 to 65 feet; a head almost entirely expended in overcoming the friction of the main.

The engine-cylinders were completely jacketed, and the steam was also taken through a jacketed reservoir between the cylinders. The jacket-water was discharged through a pipe regulated by a stop-valve and weighed. The condensers are injection-condensers with horizontal air-pumps.

The actual dimensions of cylinders, pumps, and rods were as tabulated below:

DIAMETERS AND AREAS OF CYLINDERS AND PUMPS.

		Diameter at 60° F.	Diameter at 316° F.	Area of Piston.	Area of Rod.	Effective Area.	Means.
		Inches.	Inches.	Sq. in.	Sq. in.	Sq. in.	
H. P. cylinder A.	Back . .	26.98	27.02	573.4	17.7	555.7	
" " A.	Front .	26.98	27.02	573.4	23.8	549.6	553.5
" " B.	Back . .	27.02	27.06	575.1	17.7	557.4	
" " B.	Front .	27.02	27.06	575.1	23.8	551.3	
L. P. cylinder A.	Back . .	53.99	54.07	2296.2	7.0	2289.2	
" " A.	Front .	53.99	54.07	2296.2	17.7	2278.5	2285.1
" " B.	Back . .	54.02	54.10	2298.7	7.0	2291.7	
" " B.	Front .	54.02	54.10	2298.7	17.7	2281.0	
Pump-plungers.	Back . .	39.90	. . .	1250.0	16.8	1233.2	1241.6
"	Front .	39.90	. . .	1250.0	0.	1250.0	

The boilers were single-flued Cornish boilers, four in number, each 28 feet in length and 6 feet in diameter, with a single flue 3 feet 6 inches in diameter for the greater part of the length. The length of the grate was 4 feet 6 inches. Hence the grate area of the four boilers was 60 square feet.

Measurement of the Feed.—The feed was supplied at a nearly constant temperature of 51 degrees, the ordinary feed

arrangements which supply the boilers with hot water from the jackets and hot well being disconnected. The boiler feed-pump took its steam from the boilers in use and exhausted into the tank, from which it pumped. The whole of the steam used was therefore recondensed and returned to the boilers. Of the heat supplied by the boilers to work the feed-pump nearly all was returned to the boilers. A small portion—viz., that due to the useful work of pumping and that lost by radiation from the tank—was no doubt lost, a small error telling against the main engines.

Measurement of the Air-Pump Discharge.—The air-pump discharge was led into a wooden tank with stilling screens. From this it was discharged through a sharp-edged circular orifice freely into the air. The diameter of the orifice was carefully tested after the trials, and the coefficient of discharge from similar orifices is known to be 0.599. The temperature and head were noted every 7½ minutes. The temperatures were taken by a fixed zero thermometer, with open scale, and verified at Kew.

Measurement of Length of Stroke.—As the stroke is variable, an arrangement of indicating-fingers was attached to each engine, and the length of stroke of each engine was noted every quarter of an hour.

Indicated Power.—The indicated power was taken by four Richards's indicators, which were afterwards tested, with the result that no important error was found in any part of the scale with any of the springs. Diagrams were taken every half-hour.

TRIAL OF ENGINES ON NOVEMBER 5 AND 6, 1888.

This was a 24 hours' trial, the coal consumption being measured as well as the efficiency of the engines. The trial commenced 10.22 A.M., November 5, and ended 10.22 A.M., November 6.

The barometer varied a little during the twenty-four hours, the mean being 29.78 inches (corrected), corresponding to 14.627 pounds per square inch.

The temperature of the injection varied from 48.6° Fahr. to 49.5°, the mean being 49.2° Fahr.

The mean boiler-pressure was 60.29 pounds per square inch (74.92 pounds absolute).

The mean vacuum shown by a mercury gauge on the engine was 27.76 inches, or 13.63 pounds per square inch.

The total head of water on the pumps was about 55 feet at starting and 53.5 feet at the end of the trial. It varied little during the trial, and the mean head was 53.68 feet.

The air-pressure in the compensating air-vessel varied from 118 pounds to 122 pounds per square inch above the atmosphere.

Speed and Length of Stroke.—The speed was remarkably constant, and averaged 17.282 double strokes per minute. The length of stroke varied from 42.32 inches to 43.56 inches; the mean length of stroke was 43.06 inches for engine A and 43.05 inches for engine B.

Indicated Horse-Power.—The reduction of diagrams taken every half-hour for the first eight hours, and every hour afterwards, gave the following results. The variation of the diagrams was very small.

			Indicated Horse-Power.	
Engine	A.	High-pressure, back, 31.662		
"	A.	Low-pressure, back, 31.145 = 62.807		128.668
"	A.	High-pressure, front, 34.176		
"	A.	Low-pressure, front, 31.685 = 65.861		
"	B.	High-pressure, back, 35.856		
"	B.	Low-pressure, back, 28.073 = 63.929		126.849
	B.	High-pressure, front, 35.236		
"	B.	Low-pressure, front, 27.684 = 62.920		
		Total indicated horse-power of both engines		255.517

The Pumps.—The mean lift was 53.68 feet; mean length of stroke, 3.58,9 feet. Number of strokes per minute, 17.282.

Hence the pumps lifted 13,407 gallons* per minute. The pump horse-power is 217.06, consequently the mechanical efficiency of the engines and pumps is 0.8495.

The Feed- and Jacket-Water.—The feed-water had a mean temperature of 51.07 degrees. The total feed-water used was 108,537.4 pounds, or 4522.39 pounds per hour. The amount of drainage from the jackets was 706 pounds per hour. Consequently, reckoned per indicated horse-power per hour, the quantities were:

Total feed (at 51.07 degrees) per indicated horse-power per hour	17.700
Jacket condensation	2.763
Used in the cylinders	14.937

Air-Pump Discharge.—The mean head over the orifice was 1.7033 feet, and the mean temperature 74.965 degrees. The total air-pump discharge was 2586 pounds per minute, or 2522.4 pounds of injection-water and 63.6 pounds of condensed steam.

Heat rejected by the Engine per Indicated Horse-Power per Minute.—The heat required to raise the whole air-pump discharge from 49.2 to 74.965 degrees. We get for the heat rejected 260.7 thermal units per indicated horse-power per minute. This is Donkin's coefficient. The more accurate estimate of the heat rejected is as follows:

	Thermal Units.
Heat due to 2522.4 pounds of injection-water per minute raised from 49.2° Fahr. to 74.965° Fahr.	64,990
Heat due to 63.6 pounds of feed-water raised from 51.07° Fahr. to 74.965° Fahr.	1,519
Heat due to 11.78 pounds of jacket-water raised 256.3° Fahr.	3,020
	69,529
Heat rejected per indicated horse-power per minute	272.1
Add converted into work	42.7
	314.8

Which neglects the loss by radiation.

* It will be understood that these are English gallons of 277.274 cubic inches.

Heat used, reckoned from the Boiler-Pressure.—The total heat of the steam, considered dry, reckoned from the feed-temperature at the mean boiler-pressure, is 1156.5 thermal units per pound. Consequently, the heat delivered from the boiler to the engine was 341.1 thermal units per indicated horse-power per minute. The difference between this and the previous estimate of 314.8 represents loss by radiation, error due to the presence of priming water in the steam, and errors of observation.

TABULATED RESULTS.

Double strokes per minute	17.282
Boiler-pressure	60.29 lbs. per sq. in.
Feed-water per minute	75.37 pounds.
Jacket drains per minute	11.77 "
Temperature of steam	307.36 deg. Fahr.
Pressure on pump	23.26 lbs. per sq. in.
Pressure on compensators	120 " " "
Mean pressure in high-pressure cylinders	32.92 " " "
Mean pressure in low-pressure cylinders	6.905 " " "
Temperature of injection	49.2 deg. Fahr.
Temperature of air-pump discharge	74.965 deg. Fahr.
Head over orifice	1.7033 feet.
Air-pump discharge per minute	2586 pounds.
Injection-water per minute	2522.4 "

Heat passing through engine per indicated horse-power per minute:

Thermal units from boiler in saturated steam through cylinders from feed temperature	287.8	
Latent heat of jacket-steam	41.45	
	329.25	
Heat rejected in air-pump discharge	260.24	
Converted into work	42.75	
Radiation and error	26.26	
	329.25	
Indicated horse-power	255.517	
Pump horse-power	217.06	
Mechanical efficiency	.8495	
Feed per indicated horse-power per hour through cylinders	14.937	pounds.
Feed per indicated horse-power through jackets	2.763	"
Piston-speed per minute	124	feet.

It should be noted here that the engines worked for twenty-four hours with the greatest regularity of speed and stroke, and this although the steam- and expansion-valves remained untouched after their first adjustment at starting.

THE BOILERS.

Measurement of Coal used.—The ash-pits were cleared before the trial, and afterwards nothing was removed till the end of the trial. The fires were cleaned before the trial began, and again at 4 A.M. The fires were not touched at the end of the trial, but the ash-pits were immediately cleaned, and the whole of the ashes treated thus:

First the clinkers, including those removed from the fires at 4 A.M. (six hours before the end of the trial), were separated and weighed. The rest of the ashes were sifted through a sieve with a half-inch mesh. All that passed through the sieve is treated as incombustible ash, although probably one-third of it is unburned carbon. What did not pass through the sieve is treated as unburned fuel. Analysis in similar cases has shown that the cinders retained by the sieve are almost entirely carbon.

The coal account then stands thus:

	Pounds.	Pounds.
Gross weight of coal brought into boiler-house . .		11,180
Left on the floor at the end of the trial	99	
Cinders sifted out of the ashes	132	231
Total coal used		10,949
= 456.2 pounds per hour.		
The residue consisted of clinkers	66	
Incombustible ashes	366	
	432	

The clinkers and ashes amount to 3.9 per cent. of the coal used.

The rate of combustion was 7.24 pounds of coal per square foot of grate, or 0.19 pound per square foot of heating surface per hour. The coal used per indicated horse-power per hour was 1.785 pounds,—a very good result, as the feed was supplied at 51° Fahr., and the rejected heat from the jacket-drains

was wasted. The evaporation was 9.914 pounds of water from 51.07 degrees at 307.36 degrees per pound of coal, including clinkers and ashes. This corresponds to an evaporation of 11.867 pounds per pound of coal from and at 212 degrees.

Calorimetric Value of the Coal.—The heating power of the coal has not been directly determined, but good Welsh coal is known to contain about 89 per cent. of carbon and 4 per cent. of hydrogen, the rest being oxygen, nitrogen, and ash. The calorimetric value of such a fuel is $14{,}500\ (0.89 + 4.28 \times 0.04) = 15{,}387$ thermal units per pound. But this is reckoned for a dried sample of coal, and makes no allowance for the latent heat of the steam produced in combustion. There would be produced by combustion 0.36 pound of water per pound of coal, and the latent heat of this would be 348 thermal units, so that the available heat of a pound of dry coal would be 15,039 thermal units. The coal as taken from the yard would contain at least 1 per cent. of moisture, so that the available heat of one pound of the coal as weighed and used would be:

	Thermal Units.
Heat due to 0.99 pound of coal	14,888
Less latent heat of 0.01 pound of water	10
	14,878

Available heat, 14,878 thermal units per pound of coal as weighed and used. Taking this value, the total heat due to the combustion of the coal is 26,557 thermal units per indicated horse-power per hour, or 442.6 thermal units per minute per indicated horse-power. Of this, 341.1 has been shown to be delivered to the steam. There remains 101.5 thermal units per indicated horse-power per minute to account for as losses in the boilers. The efficiency of the boilers is 0.77. The coal gave to the steam 11,466 thermal units per pound of coal used.

Anemometer Observations.—Observations at each boiler every half-hour gave the following volumes of air

entering per minute in cubic feet at the temperature 79.5 degrees of the boiler-house:

Boiler	*J*	*K*	*L*	*M*
Quantity of air in cubic feet per minute	420	438	486	360

Hence the total quantity of air used was 1704 cubic feet per minute, or 225 cubic feet per pound of coal. The weight of the air used was 7489 pounds per hour, or 16.42 pounds per pound of coal. As the coal requires nearly 12 pounds of air per pound for perfect combustion, the quantity of air used was moderate.

The mean temperature of the flue from the pyrometer observations was 422 degrees.

Tabulating the results stated, we get:

	Per Hour. Pounds.	Per Indicated Horse-Power per Hour. Pounds.
Coal used	456 2	1.785
Air used	7489.0	29.310
	7945.2	
Less ashes and clinkers	18.0	
Total weight of furnace gases	7927.2	31.03

Heat Used and Lost in the Boilers.—The thermal units of heat developed in the furnaces were applied thus:

	Thermal Units per Ind Horse-Power. Per Hour.	Per Cent.
Total heat due to coal and gas	26,557	100
Given to steam	20,466	77.1
Carried off in furnace gases	2,657	10.0
Probable loss due to opening fire-doors to stoke	265	1.0
Due to carbon in ashes	284	1 1
Radiation and unaccounted for	2,885	10.8

This calculation depends on an assumption of the calorific value of the coal, but this cannot be far wrong. It assumes that the steam supplied to the engines was dry. If there was any priming-water, the heat given to the steam would be less.

On the other hand, probably, the losses due to moisture in the coal and to air entering the furnaces during stoking are underestimated.

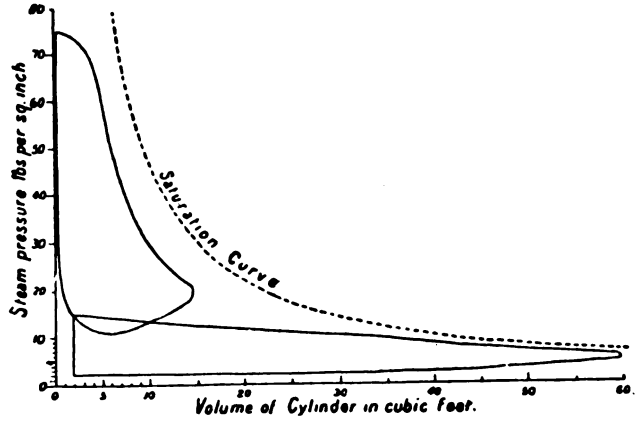
Duty of the Engines.—The work done by the engines during the 24 hours' trial was 106,010,000 foot-pounds per 112 pounds of coal.* During this trial the ordinary conditions of the engine were altered and heat rejected which is ordinarily used.

Correcting for this, the duty of the engines in normal conditions of work must be 111.5 millions according to the results of this 24 hours' trial.

To accompany this report drawings were sent, which are reproduced, as follows:

1. A mean diagram, Fig. 236, drawn from the diagrams taken on engine A at 12.30 P.M. On this has been plotted a

FIG. 236.



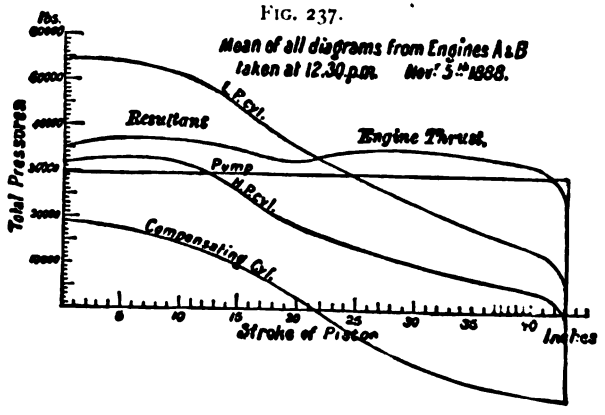
saturation curve for the mean speed per stroke during the trial. Since the indicated power varied so little, this saturation curve must be very approximately the true curve for the actual diagrams. The re-evaporation during the stroke is

* The difference between the British and the American standard of hundredweight will be noted, and the proper credit given.

very marked, as was to be expected from the large jacket condensation.

2. Mean diagrams, Fig. 237, from all the diagrams of both

FIG. 237.



engines taken at 12.30 P.M., are plotted so as to show the effective thrust of the engines at each point of the stroke. A curve of cosines is drawn, giving the ± thrust of the compensators. Combining this with the engine diagram, the resultant thrust is obtained. The effect of the inertia, however, is neglected. It will be seen that the resultant thrust is remarkably uniform, and probably the effect of the inertia of the moving pistons and plungers is to increase the uniformity of this thrust.

FIG. 238.

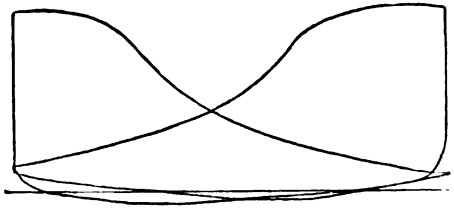
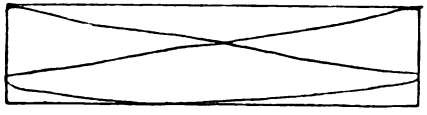


FIG. 239.



Indicator diagrams of both the high- and low-pressure cylinders are shown in Figs. 238 and 239.

CHAPTER XVIII.

HIGH-DUTY PUMPING ENGINES—DIRECT-ACTING.

THE direct-acting engine possesses features which are of especial value in handling water. As originally constructed they were wasteful in the extreme; subsequently, by the addition of compound steam-cylinders, a liberal saving of steam was secured, but with the best types of compound condensing direct-acting engines they fell far short of the duty attained by the crank and fly-wheel engines. The want of a suitable attachment to direct-acting pumping engines in order to secure the advantages of steam expansion, and thus compete with the crank and fly-wheel pumps, were recognized long ago, but realized only within the past ten years.

Cameron's high-duty engine, the invention of the late A. S. Cameron, was patented in 1876, but so far as the writer is aware has never been placed upon the market. It is probably the first device of its kind for using steam expansively in direct-acting pumps. The following engravings and description present Mr. Cameron's views upon the subject.

In Figs. 240 and 241 letter *A* designates the steam-cylinder and letter *B* the pump-cylinder, the pistons of which are connected by a rod, *C*. On this rod is secured a cross-head, *D*, which connects with the compensating-gear by two rods, *E*, and with the expansion-gear of the steam-cylinder by a rod, *F*. These rods are provided at their ends with hooks, which catch over pivots formed at the end of the cross-head, so that said rods can be readily disengaged, and the pumping engine can be worked in the ordinary manner without expansion. The compensating-gear which is represented in the drawing

consists of a cylinder, *G*, which is secured to the top of the air-vessel *H* of the pump. From one side of this air-vessel extends the ascension-pipe *I*, and from its opposite side extends a pipe, *J*, which leads into the top of the cylinder *G*. In this cylinder works a piston, *K*, from which extends a rod, *L*, through a stuffing-box in the cylinder-cover, and on the upper end of this rod is secured a yoke, *M*, which is guided between flanges *a a*, cast with or otherwise secured to the sides of the cylinder *G*. With this yoke are combined two pairs of toggle-levers, each pair consisting of two sections, *N O*, which are connected together by pivots *b*, while the upper sections are attached to the yoke *M* by pivots *c*, and the lower sections of the sides of the pump-cylinder by pivots *d*. The pivots *b*,

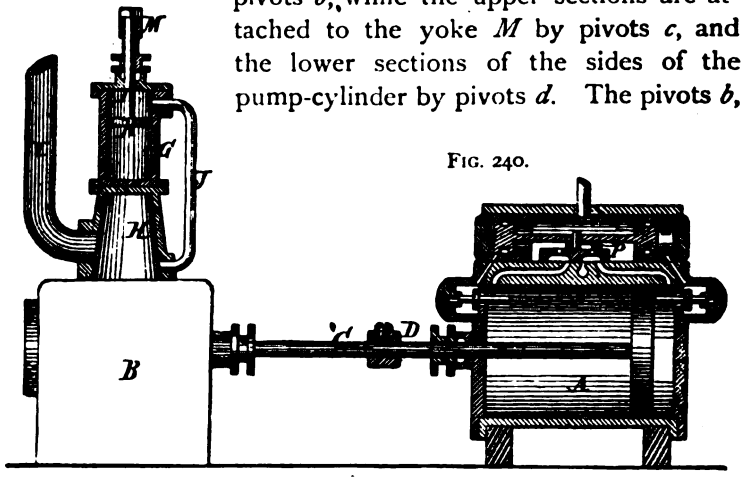


FIG. 240.

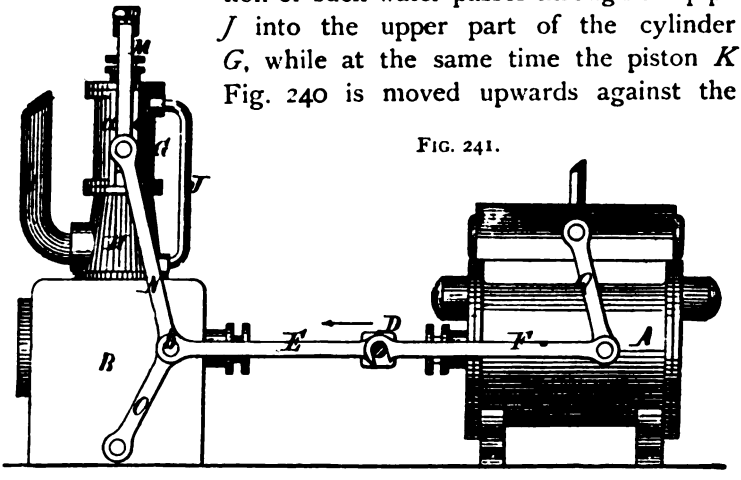
which form the connection between the sections *N O* of the toggle-levers, support the outer ends of the rods *E*, which connect the compensating-gear with the cross-head *D*.

The cut-off valve or expansion-slide *P* is moved by a lever, *Q*, which connects with the cross-head *D* by the rod *F*. The valve-gear represented in the drawing is of that class known as "steam-moved valves."

Working without Expansion.—When it is desired to work the engine without expansion, the rods *F* and *E* are dis-

connected from the cross-rod; but when the rods *F* and *E* are connected to the cross-head, as shown in Fig. 241, the operation is as follows: When the steam-piston moves in the direction of the arrow, Fig. 241, the water ejected by the pump is forced up through the ascension-pipe *I*; but a portion of such water passes through the pipe *J* into the upper part of the cylinder *G*, while at the same time the piston *K* Fig. 240 is moved upwards against the

FIG. 241.



pressure of the water acting on it. As soon as the toggle-levers *N O* pass their centre of motion, however, the piston *K* begins to descend, and as it is continually exposed to the pressure of the water acting on its upper surface, it moves down with considerable power; and if steam has been cut off at any point of the stroke of the steam-piston, the decreasing power of the expanding steam during the latter part of the stroke of the steam-piston is compensated for by the pressure of the water on the piston *K* during its descent. It will also be noticed that by the combination of the toggle-levers *N O* with the piston *K* and cross-head *D*, the power exerted by the piston *K* on said cross-head increases during its descent, while at the same time the pressure of the expanding steam on the steam-piston decreases.

Employing Weights or Springs.—It will be seen from this description that the power exerted by the compensating-

piston *K* during the second half of each stroke of the steam-cylinder changes with the resistance to be overcome; that is to say, if the height to which the water is forced increases, the downward pressure of the water on the compensating-piston, and consequently the power exerted by the compensating-gear, increases in a corresponding ratio, and *vice versa;* and for this reason, whenever applying a compensating-gear to a steam-pump, Mr. Cameron preferred to use the mechanism shown in the drawing; but, if desired, a simple weight or spring may be substituted for the piston *K*, said weight being raised during the first part of each stroke of the steam-piston, and being made to descend during the second half; but it is obvious that the compensating power exerted by such weight will be uniform no matter what may be the resistance to be overcome.

Davies's High-Duty Engines.—In 1879, Mr. J. D. Davies sought to accomplish the same result as that of Mr. Cameron, though in a somewhat different manner, as shown in Fig. 242, a device which he patented the following year.

Mr. Davies's object was to cause the piston of a steam-engine or other motor to transmit a constant and equable force throughout every portion of its stroke, and prevent any undue strains on the different parts of the engine when operated expansively, and to regulate the power to compensate for the variable resistance offered by the mechanism to be actuated.

Referring to Fig. 242, *A* represents the frame, *B* the steam-cylinder, and *C* the pump-cylinder, of a direct-acting steam-pump, which parts may be of any approved construction. *D* is the piston-rod, to the centre of which is secured a cross-head, *E*. At opposite sides of the piston-rod *D* are located the auxiliary or equalizing cylinders *F F'*, each of which is provided with trunions *a a'*, which are journaled, respectively, in the lateral bracket *G* of the engine-frame and cross-bar or frame *G'*, the latter being supported on the uprights or pillars *H H'*.

The piston-rods *I I* of the oscillating cylinders *F F'* are

provided with eyes on their outer ends, through which are inserted the bolts *L L'*, which extend through the opposite

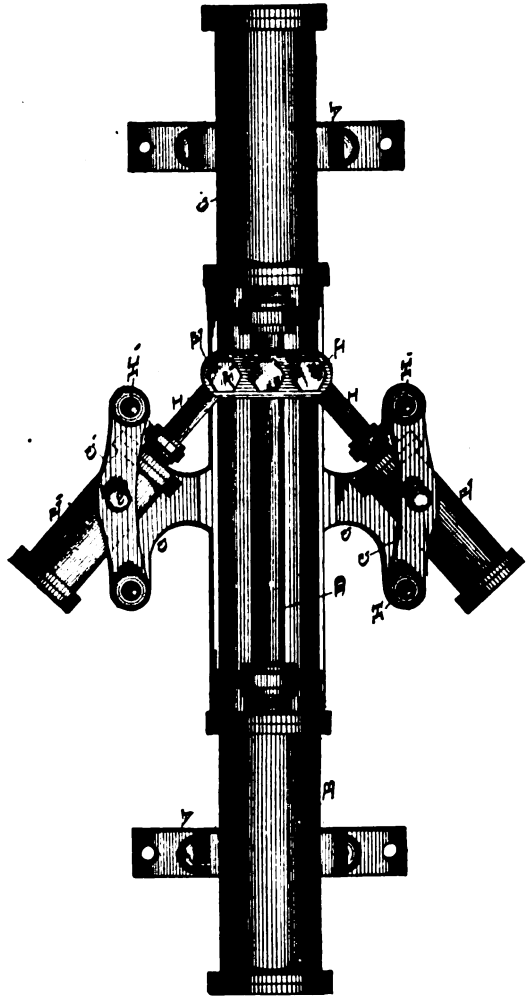


FIG. 242.

ends of the cross-head *E*, thus pivoting the pistons to the cross-heads.

The oscillating cylinders *F F'* are, in fact, single-acting oscillating engines, steam, compressed air, or liquid being ad-

mitted through the trunnions in any suitable manner to the rear portions of the cylinders to exert a constant outward pressure on the pistons within said cylinders. When the main piston is at the commencement of its stroke, the auxiliary or equalizing engines or cylinders will offer the maximum resistance to the outward movement of the main piston, as, of course, the main and auxiliary piston-rods are located more nearly in parallelism at the opposite ends of their stroke than at any other part of the stroke of the engine. As the main piston-rod moves outward the equalizer is gradually turned on its trunnions, and its resistance to the main piston gradually diminishes until the main and auxiliary piston-rods are at right angles to each other, at which point the equalizers or auxiliary cylinders will have no effect on the action of the engine. When the centre of the main piston-rod passes the first half of its stroke the equalizers or auxiliary cylinders begin to act and assist the movement of the main engine, as the power of the main steam-cylinder is then re-enforced by the power exerted upon the pistons in the auxiliary cylinders, and the power of the latter constantly increases as the cylinders are swung around and their piston-rods approximate more closely a line of movement in parallelism to that of the main piston-rod. The angles of the auxiliary piston-rods are constantly changing from the commencement to the end of the stroke of the engine or other motor to which the equalizers are attached, and hence the relative amount of resistance offered and assistance afforded by the equalizers varies throughout every portion of the stroke. The force of the equalizers may be varied throughout the different portions of the stroke of the engine or motor by a cut-off applied to any part of the supply-pipe, or by varying the cut-off or pressure on the main piston of the engine or motor, or by varying the pressure on the pistons of the equalizers during the different parts of the stroke of the main piston-rod, or by varying the proportions of the stroke of the equalizers to that of the stroke of the main piston-rod, or by changing its position during the stroke to vary the pressure, as desired.

In single-acting engines, or engines having a greater load in one direction, the equalizers may be placed nearer one end of the stroke, thus offering greater resistance in one direction and giving more assistance in the other than when located midway the stroke.

The Worthington High-Duty Engine, the invention of Mr. Davies, was purchased by the firm of Henry R. Worthington. The value of the main features of this design were fully recognized by them, but its want of completeness in order to make it valuable in practical use in pumping engines required much thought, time, and money. Mr. C. C. Worthington, of the above firm, became personally interested in working out the several devices, which, in combination with the original invention, have demonstrated fully the practicability of this method of compensation in direct-acting engines. Nor is this all,—the compensating device must be applicable to duplex engines; therefore this new train of mechanism must be applied to each engine separately, and both sets of compensators must be controlled by a common regulating device yet to be designed. A sectional elevation of this engine is shown in Fig. 243.

Adaptation to Duplex Engines.—In the case of the duplex pumping engine, its one peculiar feature is the variable and adjustable pause which its valve motion permits at the end of each stroke. The effect of this pause when dealing with the pumping of fluids is well known to be of great practical value. The combining with such an engine of a fly-wheel or any other contrivance in which the momentum of moving masses is employed for the purpose of enabling it to run under higher ratios of expansion, and consequent economy, would rob it entirely of this distinctive and valuable characteristic, because only a determinate and instantaneous pause at the end of the stroke would then be possible. This fact has hitherto prevented the highest rate of economy being attained by this type of engine. By combining with it the

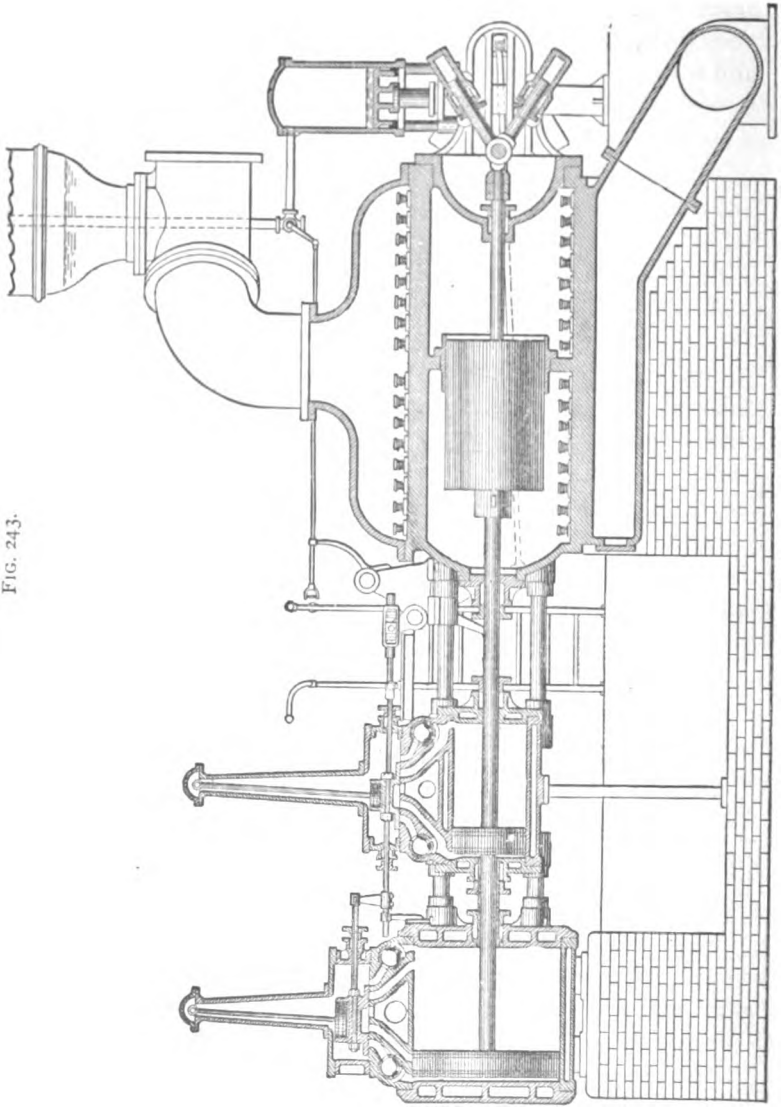


FIG. 243.

compensating-cylinders, as described, a result heretofore considered impossible with it is accomplished,—viz., not only does it realize as high ratios of expansion as are possible with any fly-wheel type, but the valuable and adjustable pause at the end of the stroke is retained. The duplex pumping engine thus becomes capable of attaining the highest economic results without the quality of its motion being affected in the least or its well-known smooth and quiet action sacrificed.

Fluid used in the Compensating-Cylinders.—The first suggestion was to use steam in the compensating-cylinders, but, upon further consideration, compressed air was thought to be better suited for that service. The first engine built was for experimental purposes only, and this engine was fitted with an air-compressor and tank with a distributing system leading to the four compensating-cylinders. The air-pressure in the tank could be fixed and maintained at any pressure best suited to the work, or conditions of service. Notwithstanding the many advantages which compressed air seemed to possess for this particular service, it was thought preferable to fill the compensating-pipes, cylinders, etc., with water, and have the compressed air act upon it, as by this means a liquid packing is interposed between the compressed air and the pistons of the compensating-cylinders and connections, and thus acts to prevent the escape of air by leakage, and also to keep the pistons well lubricated.

Safety Attachment.—Up to this point reference is had only to the compensating-gear and its usefulness as a means of using steam expansively, but a new feature was now introduced by which the steam-valves were to be operated from the cross-head of the compensating-gear. In the regular duplex engines the distribution of steam is effected by suitable links operated from the cross-head attached to the main piston-rods; but by connecting the steam-valve gear with the compensating cross-head, it is apparent that if a breakage of a rod should occur at any point between the steam-pistons

and the water-plungers, the compensating-gear for that side of the engine would be thrown out of service, and the motion of the opposite engine would cease immediately upon the completion of the then unfinished stroke. This is a feature of great importance, as, if the valves were operated so as to allow the steam to act upon the pistons after the rod was broken, the pistons, being relieved of their load, would be moved violently in the cylinders, and would be liable to cause great damage to the engine.

In ordinary engines this can be done by operating the throttle-valve so as to properly regulate the amount of steam admitted to the steam-cylinders, and, so far as the available power which is developed by the main cylinders is concerned, this can be readily done in the present engine; but as to the power developed by the compensating-cylinders, the case is different. The power developed by these cylinders, as will be readily seen, is always the same, regardless of the quantity of steam admitted to the main cylinders, and cannot be regulated or controlled by the operation of the throttle-valve. When, therefore, the engine is to be started, or run with little or no load, there must of course be sufficient steam admitted to the main cylinders to overcome the resistance offered by the pistons of the compensating-cylinders up to the middle of the stroke. As soon, however, as the middle of the stroke is reached, the main pistons are not only relieved of this resistance or load, but are assisted by the pistons of the compensating-cylinders, so that, unless means were provided for preventing it, the last half of the stroke would be made with great violence. In pumping against heavy pressure there is also, of course, some danger that the force-main may burst, which, in case it should happen, would suddenly relieve the engine of the whole or a large part of its load, thus allowing the speed of the engine to become suddenly accelerated to such an extent as to be liable to occasion damage.

The original or experimental engine had the compensating-cylinders placed between the steam- and water-ends, but at the conclusion of the experiments, which lasted several

months, the compensating-cylinders were changed from between the steam- and water-end, and placed at the outer end of the water-cylinder, as shown in Fig. 243. There is also shown immediately above the compensating-cylinders an accumulator for securing the necessary pressure in the compensating system.

The accumulator used is of the differential type; it has below a small cylinder filled with water or oil, within which its plunger moves, while above it has a larger cylinder filled with air, and within which there is a piston-head which fits closely to the cylinder, and is at the same time attached to the top of the plunger in the lower cylinder.

By this arrangement it will be seen that the pressure per square inch on the plunger or ram of the accumulator will be the pressure per square inch on the piston-head in the upper cylinder, multiplied by the difference between the area of the piston-head and the lower plunger. This difference of areas is a matter of calculation based upon the particular service for which the pump is constructed. The pressure in the air-cylinder is controlled by the pressure in the main delivery-pipe of the pump as it is connected to that pipe. This connection with the main has another very important use, as the power exerted by the compensating-cylinders is a very considerable part of the power used in driving the pump-plunger at the latter part of its stroke, and it will be seen that if, for any cause, either by the breaking of the main or otherwise, the load is entirely thrown off the pump; the plunger cannot make a disastrous plunge forward, for the reason that the steam in the steam-cylinder is, by reason of its expansion, too low in pressure to drive it, while the fall of pressure in the main has robbed the accumulating-cylinders of their power.

The practical working of the completed machine may be illustrated graphically as follows: The indicator diagrams, Fig. 244, are from an engine of this type, the ratio of

FIG. 244.

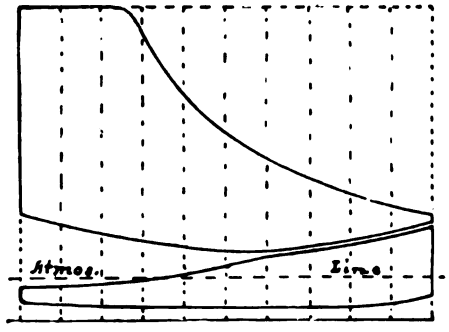
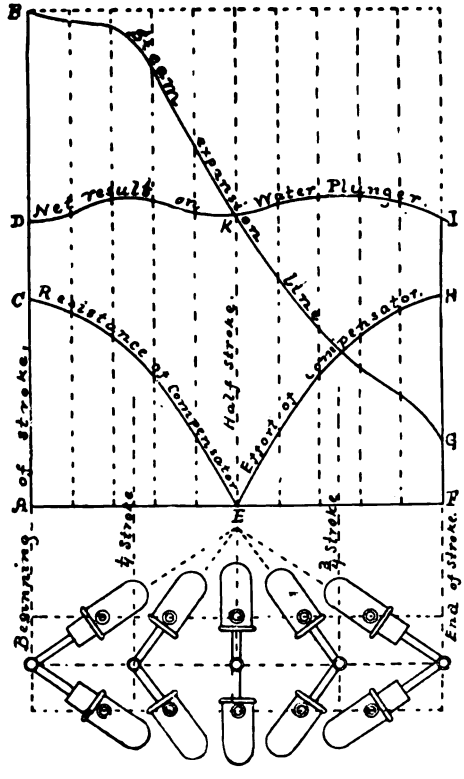


FIG. 245.



high- to low-pressure cylinders being as 1 to 4 in area. Retaining the high-pressure lines, and multiplying the vertical low-pressure lines by 4 (the difference in areas), there is had as a resultant the line of steam-effort shown in Fig. 245. Immediately underneath this latter illustration is a sketch showing five positions of the compensating-cylinders, corresponding to each quarter of the stroke from the beginning. Curved lines are drawn in this diagram, Fig. 245, by one of which, in the lower left-hand corner, is shown the resistance of the compensating-cylinders to the forward piston-movement for the first half of the stroke, and in the lower right-hand corner is another curved line showing the effort of the compensating-

cylinders in assisting the main piston's movement on the last half of the stroke. By deducting the resistances in the first half of the stroke, the steam line *A B* is effective only in *A D*.

In the middle of the stroke the effort of the compensating-cylinders is *nil*, so that the resistance *A C* at the beginning of the stroke is 0 at half-stroke, or at the point *E* in the diagram; but immediately on passing the centre of the stroke the effort of the compensating-cylinders gradually increases, as shown in the curved line *E H*, so that the steam-pressure *F G* at the end of the stroke is augmented by the compensator-effort *F H*, which two combined efforts give the terminal pressure *F I*, which is practically the same as at the beginning of the stroke. The combined effort on the water-plunger throughout the whole stroke, for the several ordinates in the diagram, is shown in the line *D I*.

As a proof of the remarkable manner in which the compensating-cylinders do compensate for the irregular pressure of the steam as thus used, if a measurement be taken between the upper or steam curved line, and the lower, or compensating curved line, on any of the ordinates, in any part of the stroke, it will be found that the distance between these lines, which is the measure of the available power for driving the pump-plunger, is exactly the same as is the distance between the upper and lower lines of the water-card on any ordinate, in any part of the stroke of the pump. In fact, it may be said that the lines of power, as well as the lines of resistance, are practically parallel, and thus it is that there is exerted at all times just power enough and no more to force the water column along so quietly, so steadily, that on engines of this construction it has been found there is no possible use for an air-chamber on the delivery-mains; and thus is the apparently paradoxical problem solved of producing a perfectly uniform motion and pressure in a steam-pump using steam-power variable to the last degree, and without the use of shafts, cranks, fly-wheels, or heavy vibrating beams.

THE GROSHON HIGH-DUTY ENGINE.

This is a recent invention by Mr. John A. Groshon, of New York City, and patented some three years ago. The invention has for its object the economic use of steam in a direct-acting pumping engine, and for the few machines thus far constructed it has, under test, demonstrated its fitness to take rank as a high-duty engine.

A perspective view of a compound duplex pumping engine, fitted with Mr. Groshon's valve-gear and compensating-device, is shown in Fig. 246. The high- and low-pressure cylinders are arranged tandem on a bed-plate. The steam- and exhaust-valves are modifications of the Corliss type, and are so disposed in the cylinder-casting as to permit the employment of separate steam- and exhaust-ports, thus retaining the advantages of the five-ported cylinder. The steam-valves have a positive movement at the beginning of the stroke, and are released at the proper point of cut-off by an hydraulic regulator controlled by the water-pressure in the force-chamber, or delivery-main. Dash-pots are provided as in other types of releasing-gear.

The compensating-device is arranged on a girder frame of I-beam form between the steam- and water-cylinders; the compensating-cylinders are located in the pit below the engine-room floor. These latter cylinders are in communication with the main discharge-pipe from the pumps, and subject to the pressure of the same, either with or without multiplying device. Fig. 247 is a side elevation of this engine, showing the hydraulic regulator and its connections, together with the compensating-cylinder.

The levers and their connections being in the position illustrated in Fig. 248, it will be observed that the pressure within the cylinders *S* will, through the medium of the rods *P*, levers *M*, rods *J*, and levers *E*, operate to oppose the movement of the piston-rod *A* during the first portions of its stroke,

FIG. 246.

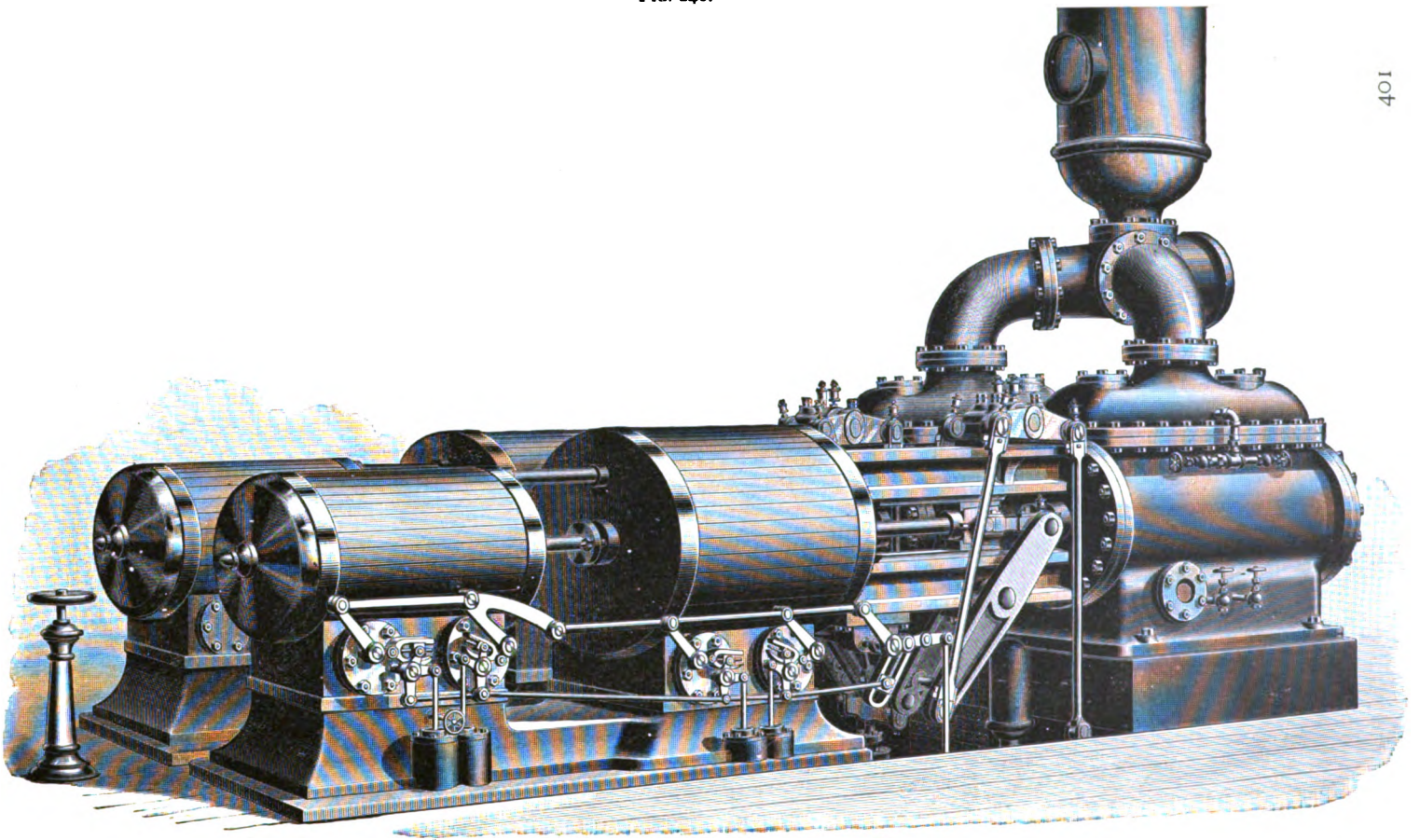
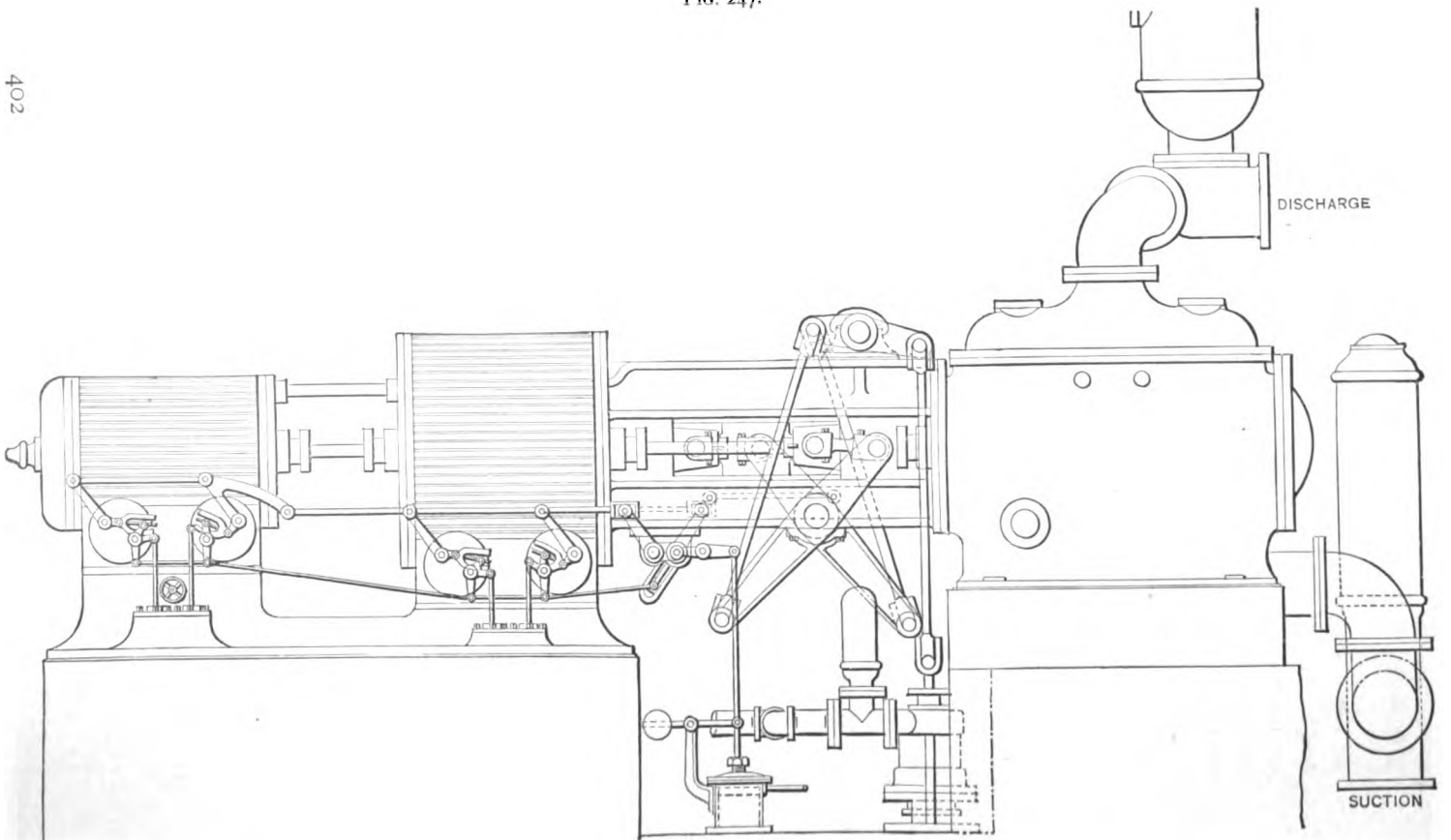


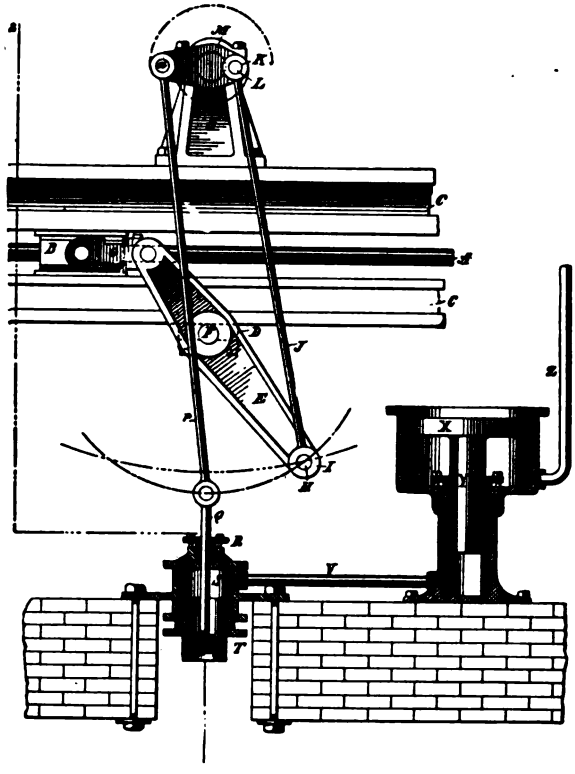
FIG. 247.



or until the levers *E* have turned sufficiently to come at right angles to the piston-rod *A*, after which the pressure within the cylinders *S*, acting on the pistons *T*, will, through the system of levers and rods, operate to assist the movement of the said main piston-rod *A*.

There are indicated by the dotted lines in the drawings the movements of the levers *M* and levers *E*, and also the extent

FIG. 248.



of the movement of the piston-rod *Q* while under pressure to permit the action of the rods *J* under the force applied to the longer arms of the levers *M*. It will be understood that during the first portion of the stroke of the main piston-rod the

longer arms of the levers *E* will, through the rods *J*, cause the levers *M* to rotate and bring their longer arms, with the rods *P*, upwards in the direction of the arrow or dotted line at the upper portion of Fig. 248, the effect being to draw the pistons *T* against the pressure within the cylinders *S*, and thereby to transmit said pressure to the piston-rod *A* in a line opposite to its movement during the first portion of its stroke; and it will also be noted, upon following the movement of said levers and rods, that after the levers *E* have passed their centre the pressure within the cylinders *S* will be transmitted to the main piston-rod *A* in line with its movement, and thereby assist the same during the latter portion of its stroke.

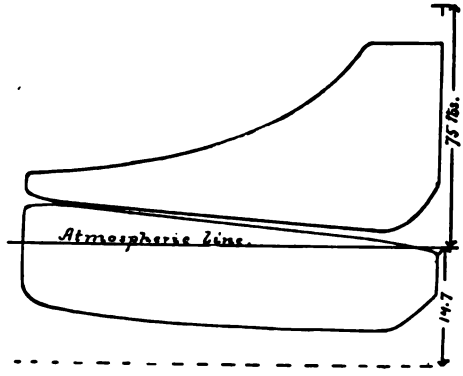
Dotted lines in the drawings illustrate arcs of two circles differing in diameter, one being long and flat, and the other of a greater curve. These indicate the lines of travel of the lower end of the rocking-lever *E* and of the rod *J*. If the lever *E* and rod *J* were allowed their natural movement from their points of suspension, they would travel on the arcs indicated by dotted lines in the drawings, and hence in operation there must be either a force applied or the overcoming of a force to compensate for the difference between these two arcs, and this force is that which is transmitted to the piston-rod in line with its length.

In the moving of the lower end of the lever *E* on its line of travel it is retarded by the rod *J*, which has a different line of travel, and to overcome this the piston in the auxiliary cylinder is compelled to rise and overcome the force from the accumulator to an extent which would turn the auxiliary lever *M* sufficiently to permit the lower end of the lever *E* and the lower end of the rod *J* to travel on the same arc, although hung from different centres, and if left in their natural state would travel on different arcs.

The indicator diagrams shown in Fig. 249 are from an engine having 12-inch high-pressure steam-cylinders, 24-inch low-pressure cylinders, water-plungers 18½ inches diameter, all of 18 inches stroke, using 75 pounds steam-pressure, and

pumping against 70 pounds water-pressure. The scale of the upper diagram before reduction was 30 per inch, and that of the lower diagram was 12 per inch. These cards show a rate of steam expansion which promises well, other things being equal, for the future of this engine.

FIG. 249.



The Davey High-Duty Engine. — Mr. Henry Davey read a paper before the British Association (Section G) in 1887, entitled "Expansive Working in Direct-Acting Pumping Engines." The author commenced by referring to a paper previously read before the British Association, in which the expansive method of working steam-pumps can be secured without the aid of a fly-wheel. The object of the paper under notice was to describe a more recent invention of his by means of which a far greater degree of expansion is made possible.

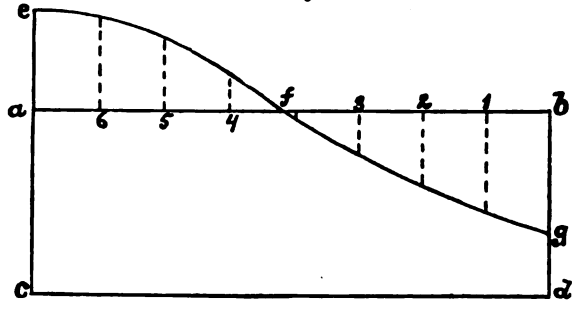
When an engine has heavy reciprocating parts, such as long pump-rods or loaded plungers, expansive working is possible because of the inertia at the beginning and the momentum towards the end of the stroke, expressed by the formula $\frac{W V^2}{2g}$.

By this method, for a considerable degree of expansion, a very high velocity must be given to the mass when the weight is small. In engines which have not long pump-rods it is not always convenient to provide weights sufficiently heavy to enable a high degree of expansion to be employed. The mechanism which Mr. Davey described equates the engine-power and pump-resistance, by causing decreasing pressure of the expanding steam on the piston of the engine to bring

a nearly constant force to bear on the pump throughout the stroke. By means of the annexed diagrams the action of the pump is made clear.

The pump resistance is represented in Fig. 250 by the parallelogram *c*, *a*, *b*, *d*, and the engine-power diagram by the figure *c*, *e*, *f*, *g*, *d*; and supposing the parts of the engine to have no weight, then means are required by which the piston

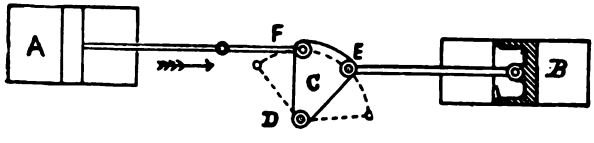
FIG. 250.



of the engine may move with varying velocities relative to that of the pump-piston, exceeding the mean velocity by the ordinates 1, 2, 3, and falling short of that velocity by the ordinates 4, 5, 6.

Decreasing Velocity of Pump-Piston.—In Fig. 251, *A* is the engine, *B* the pump-piston, and *C* a triangular frame turning on the fulcrum *D*. The pump-piston is at-

FIG. 251.

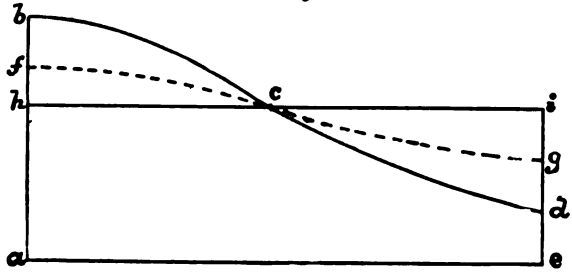


tached to the frame at the point *E* by means of a vibrating connecting-rod, and the engine-piston to the point *F* by means of a similar rod. While the engine is making its stroke in

the direction of the arrow, the pump-piston is decreasing in velocity relative to that of the engine-piston, the ratio being determined by the relative positions of *E* and *F*.

Ratio of Expansion.—In applying this mechanism to pumping engines it is first necessary to determine the ratio of expansion to be employed, and then to see how nearly the force and resistance can be equated. In Fig. 252, *a*, *b*, *c*, *d*, *e* are the combined diagrams of a compound engine working

FIG. 252.



with the given ratio of expansion *a*, *f*, *g*, *e*, the diagram of effects of the varying velocities of the engine and pump-pistons, and *a*, *h*, *i*, *e*, the pump-resistance diagram. Then acceleration of velocity takes place from *h c*, and, knowing the weight of the moving parts, the acceleration may be calculated. It will at once be seen that the mechanical advantage obtained by this mechanism greatly reduces the acceleration for a given mass.

Clearance.—In answering the discussion which followed the reading of this paper, and speaking on the variation in length of stroke, Mr. Davey said that all direct-acting engines have the stroke controlled by the action of valve-gear, and depend on the adjustment of the gear for clearance. The engine described might have 8 feet to 10 feet length of stroke, and work safely to 1 inch of the covers, or in special cases even less; but that was a fair allowance. The Cornish engine had a higher economy than any engine working with the same ratio of expansion, but the clearance there would be 2

inches to 3 inches. The economy in such cases had always been a puzzle to him. With a pressure of 30 pounds to 40 pounds and four expansions, there would be an equal effect to a rotative engine working with eight expansions and a pressure three or four times as great. Rotative engines were made for pumping under the sentimental notion that they were more economical on account of less clearance. But it must be remembered that a rotative engine must be designed to meet the worst condition of work, and therefore could not, from the nature of their design, take the full advantage, that might otherwise be obtained, of more favorable conditions. A rotative engine must depend for its expansion on the number of revolutions, and mining engineers, to take one instance, must work sometimes at slow speeds, and the energy of the fly-wheel was not then enough to work with a large degree of expansion. For this reason rotative mining engines do not in practice always work expansively; though direct-acting pumping engines are always at their maximum expansion, whatever speed they may be working at.

Fielding's High-Duty Engine.—The publication of Mr. Davey's paper was the occasion of a letter addressed to *Engineering* by Mr. John Fielding, in which he says that he does not doubt that the arrangement described by Mr. Davey as "a recent invention of the author" was so described in good faith, and without the knowledge that it had been previously invented. Nevertheless, a search through the patent records would have convinced Mr. Davey that Fielding's patent, No. 788, of 1874, most clearly anticipated him. Mr. Fielding enclosed a tracing showing the action of his compensating-gear as applied in one form of pumping engine, which is reproduced in Figs. 253 and 254.

Mr. Fielding's object was to enable engines to be worked expansively (without necessitating heavy masses to be put in motion), for the purpose of absorbing surplus energy during the first part and of giving it out during the last part of the stroke, with the end of obtaining an approximately constant

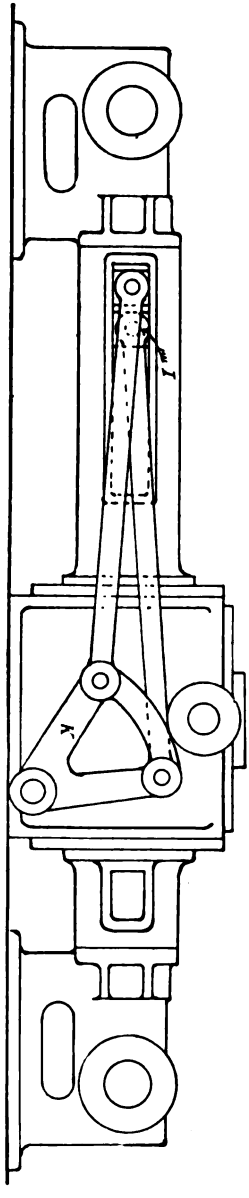


FIG. 254.

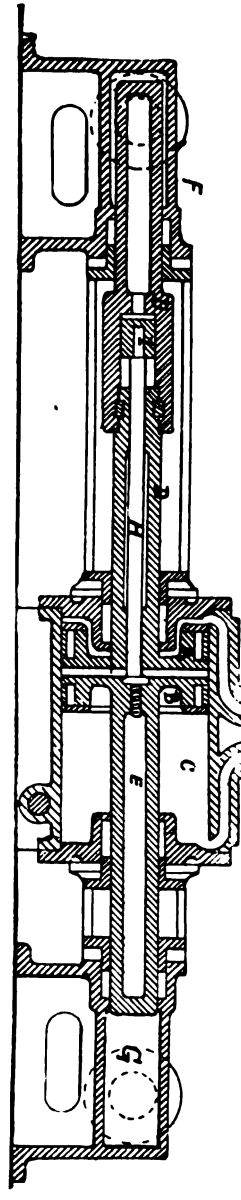


FIG. 253.

pressure on the pump-piston throughout the whole of the stroke, while having a varying pressure on the steam-piston, resulting from the expansion of the steam after the supply had been cut off.

Fig. 253 shows a longitudinal section and Fig. 254 a side elevation of one form of Mr. Fielding's invention. Two pistons, *A B*, work within the cylinder *C*, and attached to these are two pump-rams, *D E*, working through stuffing-boxes in the cylinder-covers into pumps *F G*. A rod, *H*, connecting to piston *B*, extends through centre of ram *D*, terminating in a cross-head, *I*, working within a slot formed in ram *D*.

A similar cross-head, *J*, is formed on ram *D*. *I* and *J* are connected by side-rods to quadrants *K K* in such a manner that piston *A* drives ram *E*, while *B* drives *D*, and the quadrants *K K* are so proportioned that at the commencement of a stroke the speed of the piston will be relatively smaller than that of the ram, say about one-half; at half-way the speeds will be equal, while at the end of the stroke the speed of the piston will be double that of the ram.

These relative speeds correspond roughly to the proportions shown on tracing, but the levers may be proportioned to suit various degrees of expansion. The drawing shows a single cylinder only; the compound principle would greatly reduce the variations in the pressure on the steam-pistons, and so reduce the difficulty of equalizing them.

Mr. Fielding's patent, not having been kept up, is void, and the arrangement is therefore public property. Mr. Fielding takes occasion to say in his communication, that, compared with the ingenious compensating motion introduced by the Worthington Pumping Engine Company, his had the advantage of directness and simplicity, seeing that it involves the use of no extra cylinders, stuffing-boxes, or reservoirs.

CHAPTER XIX.

HIGH-DUTY PUMPING ENGINES—FLY-WHEEL

THE high degree of economy attained by some engines of this type over ordinary direct-acting, and the general run of crank and fly-wheel pumps, is due to the design and construction of the steam-engine portion of the machine; the use of a high initial steam-pressure, together with the advantages which may be secured by a liberal steam-expansion in two or more cylinders, may be accepted as the real reason for the high degree of efficiency which engines of this type have developed.

A comparative diagram of steam-expansions has already been given, together with a simple explanation so far as relates to steam-pressures; the consideration of the steam-engine as a heat-engine is not entered upon, it being quite foreign to our present purpose, which is more especially with water-end design than with that of the steam-end.

The degree of refinement to which steam-engine design has been carried seems to leave but little more to be accomplished in that direction.

The enormous loss often referred to in recounting the difference between the known energy of coal and the net result of the best type of steam-engine is, in the present state of the art, wholly outside of the province of the engineer to correct. The heat rejected by the furnace cannot, except in minor quantity, be economically recovered, and must, therefore, be considered as one of the unavoidable losses which accompany the generation of steam. Mr. J. T. Henthorn contributed, in Vol. XII., "Transactions of the American Society of Mechanical-Engineers," the results of a test of a triple-

expansion engine, showing an indicated horse-power for each 12.94 pounds of steam per hour, the engine developing 532 H. P. In the discussion which followed the reading of this paper, Mr. George H. Barrus, the efficient chairman of the Duty-Trial Committee of the above society, who recommends that the economy of a pumping engine be referred to heat-units rather than to coal and water consumption, worked out these quantities roughly during the meeting, in which he made the number of thermal units consumed per I. H. P. per hour 14.038, and the quantity consumed per minute 234. This may be taken as representing the best modern practice in steam-engine performance; some tests recording as low as 12.6 pounds of steam per I. H. P. per hour are reported, but taken altogether, 13 pounds may be said to fairly represent the best steam economy at this time.

The simple fact that a water-end is operated by an engine having an automatic or adjustable cut-off does not make it a high-duty engine. There are pumping engines thus equipped now in use in water-works pumping stations in which the average duty is no greater than could be had by the employment of a simpler and cheaper type of compound direct-acting engine.

A high-duty engine must be designed for the particular service required of it, so that the adaptation of the steam to the water-end shall be one favorable to high economy. The steam-pressure usually determines the number of cylinders to be used. For steam-pressures averaging 100 pounds the engine would be compound, the engine having one high and one low-pressure cylinder, or one high and two low-pressure cylinders; the latter is sometimes employed in three-cylinder pumps. For steam-pressures averaging 150 pounds, the engine would be arranged for triple expansion, the work in the three cylinders being distributed approximately, thus: the 150 pounds (total pressure) in the first cylinder, 60 pounds initial pressure in the intermediate cylinder, the third or low-pressure cylinder having an initial pressure of 20 pounds, exhausting against a back-pressure of say 3 pounds above

the vacuum; this would make a fair distribution of temperatures, and would be productive of good results if properly applied.

Three pumping engines have been selected for illustration,—viz.:

1. A single, vertical, bucket- and plunger-pump, designed by Mr. E. D. Leavitt, Jr., and erected at Lynn, Mass.

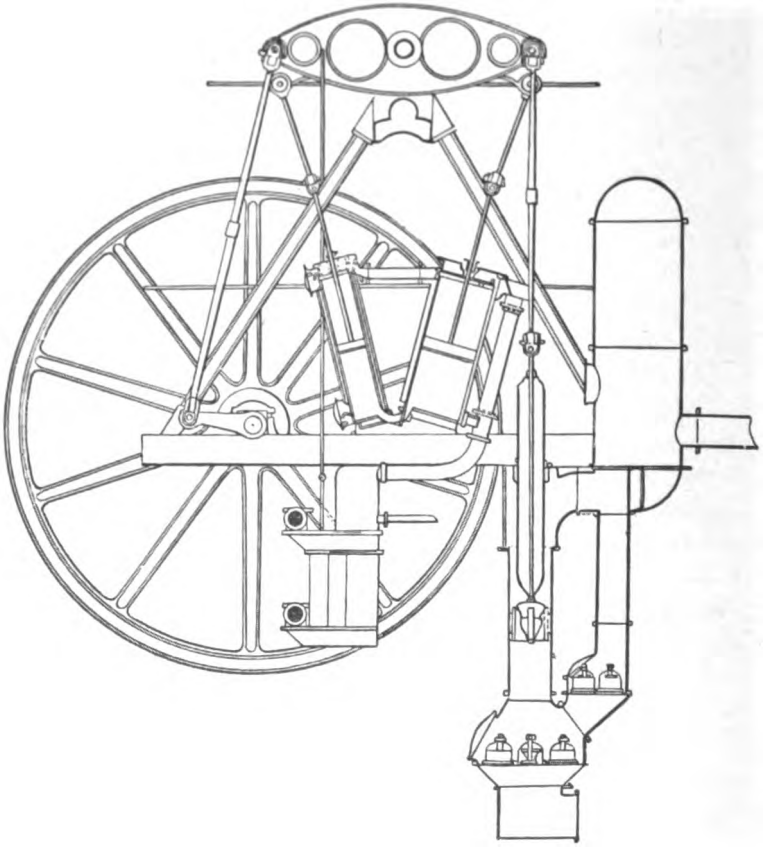
2. A pair of horizontal, double-acting plunger-pumps, designed by Mr. George H. Corliss, and erected at Pawtucket, R. I.

3. A vertical, triple expansion, single-acting plunger-pump, designed by Mr. Edwin Reynolds, and erected at Chicago, Ill.

The Leavitt Compound Pumping Engine.—A compound rotative pumping engine, designed by Mr. E. D. Leavitt, Jr., for the city of Lynn, Mass., is shown in sectional elevation in Fig. 255. In this engine the steam-cylinders are placed very nearly under the main centre of the beam, and are inclined outwardly, at the proper angle to connect with the dependent projections cast on the beam, the low-pressure cylinder being connected in the pump-end of the beam, and the high-pressure to the opposite or crank-end. The pistons thus have opposite movements; and the connections between the cylinders, for carrying the steam from the high-pressure to the low, are very short and direct. The strain on the beam is, by this construction, reduced to a minimum. The top of the high-pressure cylinder exhausts into the top of the low-pressure cylinder, and the bottom of the high-pressure cylinder exhausts into the bottom of the low-pressure cylinder. A single valve controls the connection between the two cylinders at the bottom, while two valves are used in the connection at the top, one close to each cylinder, so that the capacity of this passage is not added to the clearance of the high-pressure cylinder when being filled with steam, nor is it exhausted when the low-pressure cylinder is being exhausted. It thus an-

swers, in some measure, to the reservoir of a compound marine engine. Both cylinders are steam-jacketed at sides and ends, and, together with the steam-connections, are clothed with asbestos covering, and lagged with black walnut.

FIG. 255.



In general appearance the high-pressure cylinder approximates the diameter of the low-pressure cylinder; this is due to the fact that the lagging has been kept away from the body of the cylinder, an air-space being thus formed between the two.

All the steam-valves are of the gridiron form, and are moved by revolving cams, which have grooves of the proper shape cut in their sides, in which run steel rollers on pins projecting from the sides of rocker-arms. Those for the high-pressure steam-valves are composed of two disks, fitted closely together, but not attached. One of these disks, for the opening movement, is keyed fast to the cam-shaft, and has its outer edge of the proper shape to open the valve, so as to give it the required lead; the other, for the closing movement, has on its inner side a projecting lip, whose interior edge has the proper shape to move the rocker-arm so as to close the valve. This latter plate is not keyed rigidly to the cam-shaft, but is driven by a double feather working in a slot cut in its boss parallel with the bore, and also working in a slot cut in the shaft at an angle with the axis, so that any motion given to the feather in the line of shaft changes the relative position of the cam with its shaft, and consequently with the crank, thereby causing steam to be cut off earlier or later. The motion is imparted to the feather by the governor.

The air-pump is double-acting, and worked by a connection with the beam. The discharge from the hot well is into the pump-well, and the boiler feed-water is drawn from the hot well by an independent steam-pump.

The pump is of the Thames Ditton variety, bucket and plunger type. There is a supplementary pipe and valves to reduce the friction incident to the passing of the water through a single bucket-valve in the piston. The valves are of the usual double-beat form; the bottom, or foot-valves, are seven in number, the upper valves three,—one in the piston and two in the supplementary passage; each of the exterior bottom valves has an independent cover on the chamber for the readier inspection and removal of the valves. To supply air to the air-chamber there is a small air-cock on the lower valve-chamber. The air drawn in through this cock on the up-stroke of the pump somewhat reduces the quantity of water discharged by the pump. Careful experiments, pre-

vious to and in the early part of the test, established the effect of this cock by measurements at the weir, and a uniform opening of this cock was kept during forty-eight hours of the trial. During the other four hours the average loss of action was a trifle less than during the forty-eight hours, during one hour the cock being completely closed.

The water is drawn by the pump from a large well supplied by a main from the collecting-pond, and discharged through a 20-inch force-main. This main is of cast iron for a portion of its length, thence by a cement-lined sheet-iron pipe of the same diameter to the bottom of a gate-house in the service-reservoir. During the experiment it was thence conducted by a vertical cast-iron pipe of sufficient height to discharge into a wooden trunk, connected with a weir-box, where the capacity of the pump was determined.

The principal dimensions are shown in the following table:

Diameter of high-pressure cylinder	17½ inches.
" low-pressure cylinder	36 "
" high-pressure piston-rods	3 "
" low-pressure piston-rods	3¾ "
" air-pump	11¼ "
" pump-barrel	26 1/16 "
" plunger	18½ "
" bottom and supplementary valves, outside lower seat	15⅞ "
" bottom and supplementary valves, inside upper seat	10 "
" piston-valve, outside lower seat	22 "
" " inside upper seat	16½ "
" fly-wheel	26½ feet.
Length of stroke of steam- and water-pistons	7 "
" " air-pump	44½ inches.
Distance between end-centres of the beam	11 feet.
Steam-lead, high-pressure cylinder, top	0
" " " bottom	0
" low-pressure cylinder, top	0
" " " bottom	1/16 inch.
Exhaust-lead, high-pressure cylinder, top	1/16 "
" " " bottom	1/16 "

Exhaust-lead, low-pressure cylinder, top	$5\frac{3}{16}$ inches.
" " " bottom	3 "
Closure before end of the stroke:	
High-pressure cylinder, top	$9\frac{7}{8}$ "
" " bottom	$14\frac{11}{16}$ "
Low-pressure cylinder, top	6 "
" " bottom	$5\frac{5}{16}$ "
Clearances:	
High-pressure cylinder, top	.0231 of vol. of cylinder.
" " bottom	.0360 " "
Low-pressure cylinder, top	.0109 " "
" " bottom	.0104 " "
Top connection between cylinders	.0134 of high-press. cyl.
Weights:	
Fly-wheel	10.7 tons.
Beams	4.2 "
Moving parts connected with beams	5 "
Force-main:	
Length from pump to top of vertical pipe at reservoir	1904 feet.
Height of top of vertical pipe above bottom of well	163.34 "

The boilers are two plain horizontal tubular set in brick-work, fired beneath, flame passing the length of the boiler and returning through a number of 3-inch tubes to a front connection, thence passing up into a single flue on the top of the boilers in brick-work, and to the chimney. The principal dimensions are:

Diameter of shell	5 feet.
" steam-drum	3 "
" tubes	3 inches.
Length of shell	16 feet.
Height of drum	6 "
Length of grate	5 "
Width of grate	5 " 6 inches.

The duration of the trial was fifty-two hours, without a stop. The engine at the conclusion of the test continued its work, the fires and water in the boilers being kept as uniform as possible. The capacity of the pump was established by weir measurements at the reservoir. The actual delivery was

found to be 96 per cent. of the calculated capacity of the pump. The capacity of the full pump is 194.37 gallons; the delivery per revolution, as determined by weir measurement, was 186.55 gallons; the number of revolutions made in the 52 hours was 57,357; and the total quantity of water delivered at the reservoir during that time was 10,700,163 gallons, —or the delivery was at the rate of 205,772 gallons per hour. The delivery required by the terms of the contract was 200,000 gallons per hour.

The duty was established as follows: The area of the pump, ascertained by direct measurement, was 534.53 square inches. The load was determined from a gauge on the main, near the pump, the average reading of which was 64.2 pounds; the difference of level between centre of gauge and surface of water in pump-well reduced to pounds = 8.21 pounds; allowance for friction and bends of main, between gauge and pump-well, 1 pound; total pressure per square inch, 73.41 pounds. The length of the stroke, as measured, was 7 feet.

From the above data and the number of revolutions the work done at the pump was determined,—no deductions having been made for loss of action. The coal was picked Lackawanna coal, and the contractor was allowed to make use of any coal from the cinder that he deemed of value as combustible, but no credit was allowed for screenings or unconsumed coal. The coal was charged against the machine in gross, and the total quantity fed into the furnaces during the 52 hours was 15,160 pounds. The division of the foot-pounds of work by the weight of coal determined the duty, and the result was 103,923,215 foot-pounds of work for every 100 pounds of coal fed into the furnaces.

The Corliss compound pumping engine, erected at Pawtucket, R. I., designed by George H. Corliss, is shown in sectional elevation in Fig. 256. It presents several novelties

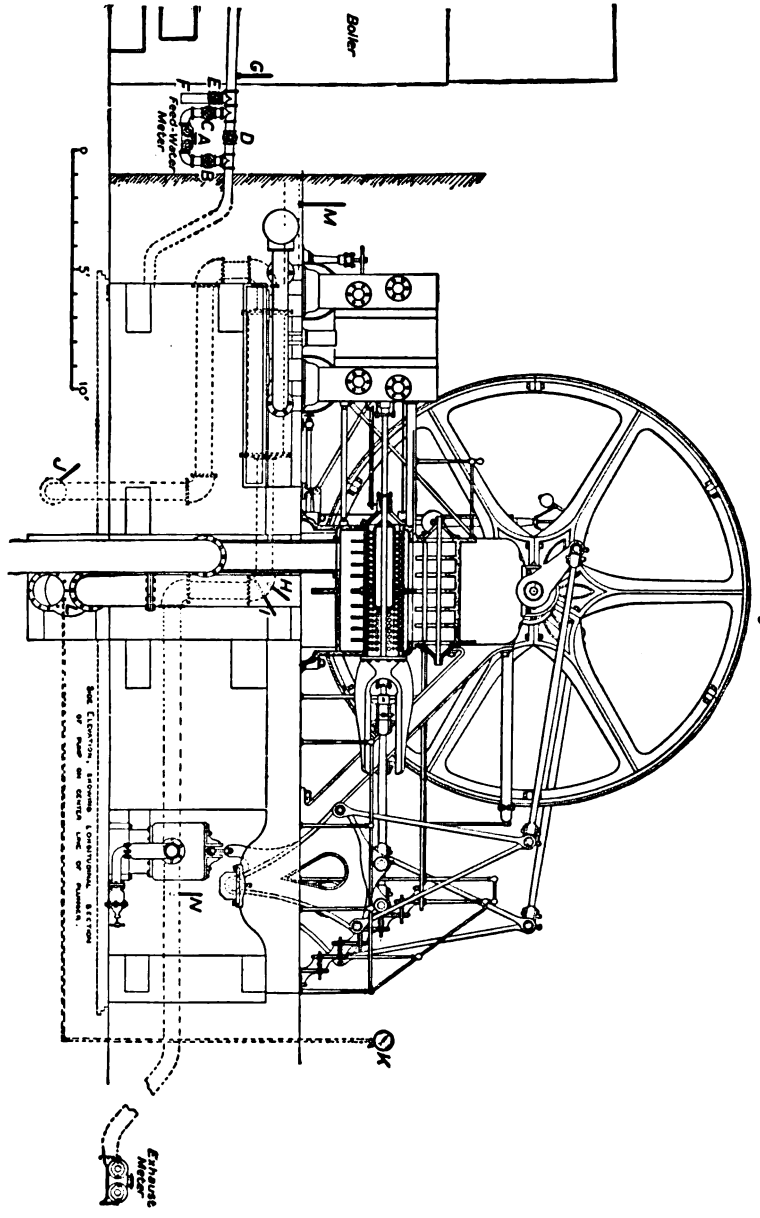


FIG. 256.

in pumping engine design that a less fearless man than Mr. Corliss would have hesitated to employ, but which fourteen years of daily use show to have been judiciously chosen for an engine of the size selected for their water-supply, which was the delivery of 3,000,000 gallons of water in 24 hours against a head of about 270 feet.

This engine consists of one high- and one low-pressure cylinder, placed side by side horizontally, with piston-rods connected directly to double-acting water-plungers. The plunger-rods are keyed to cross-heads, from which links run to vertical beams; the link connection being made in the centre of the beam and the fulcrum at the lower end. From the upper end connecting-rods are coupled to cranks, which are placed at right angles, and have a throw equal to twice the stroke of the steam-piston and plungers. The bearings for the crank-shaft are carried on top of the pump air-chambers, and braces run from these air-chambers to the main centre pedestals of the beams.

The steam-cylinders, valve-boxes, and cylinder-heads are all thoroughly steam-jacketed; steam of full boiler-pressure is used in each. The induction-valves of both cylinders are connected to one lever for hand adjustment, and are placed under the control of a governor, which limits the velocity of the engine to 52 revolutions per minute.

The steam-valves and valve mechanism are in all their salient features similar to Mr. Corliss's previous practice, but with certain modifications which were first used in the engine which furnished the motive-power of the machinery-hall at the Centennial Exhibition.

Receiver.—There is a receiver between the steam-cylinders approximately equal in volume to that of the low-pressure cylinder; the condensed water from the steam-jacket is returned to the boiler-feed, and that from the receiver, after being re-evaporated into superheated steam, is discharged into this same receiver with the effect, not infrequently, that more steam can be accounted for on the low-pressure indicator-card

than came out of the high-pressure cylinder. The re-evaporation of the condensed water is performed in a coil, placed in the boiler-flue.

A jet-condenser is employed, the water of condensation being supplied automatically from the pump-well.

The engine is adapted for the use of steam of 125 pounds pressure, which is first admitted to the high-pressure cylinder, and from which it is exhausted into a receiver, and from thence passes into the low-pressure cylinder, from which it is exhausted into a jet-condenser; the steam being expanded about twenty times its original volume at the moment of final release from the low-pressure cylinder.

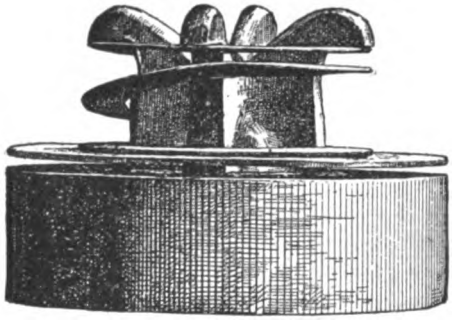
The principal dimensions of the engines are:

Diameter of high-pressure cylinder	15	inches.
" low-pressure cylinder	30⅛	"
Length of stroke, both cylinders	30	"
" cut-off, high-pressure cylinder	0.22	per cent.
" " low-pressure cylinder	0.335	"
Clearance, high-pressure cylinder	4	"
" low-pressure cylinder	3.7	"
Port area in per cent. of piston area:		
Admission high-pressure cylinder	9.04	"
Exhaust high-pressure cylinder	14.8	"
Admission low-pressure cylinder	5.04	"
Exhaust low-pressure cylinder	9.08	"
Diameter of all pump-plungers	10.52	inches.
" all rods	2¾	"

The pump-valves are similar to one shown in the full-size illustration, Fig. 257. The disk of copper forming the valve is shown lifted off of the valve-seat $\frac{1}{16}$ inch, about the extent which the valves are believed to lift during the working of the pump. The area exposed to pressure tending to open the valve is 1.8 square inches. There are 70 valves in each section of the pump. A lift of $\frac{1}{16}$ of an inch, therefore, offers an area of outlet equal to half the cross-section of the pump-plunger. The weight of the moving parts of each valve is 0.8 ounce.

To lift a valve $\frac{1}{16}$ of an inch, and maintain it against the compressive force of the spring, requires a water-pressure of ¼ pound per square inch. To overcome the inertia of the valve would require less than ½ pound of water-pressure, supposing the valve to open during the first fifth of an inch of piston-travel, or during $\frac{1}{225}$ of a second of time.

FIG. 257.



The boilers are of the upright tubular variety, three in number, also designed by Mr. Corliss, each boiler being 4 feet in diameter and 14 feet high, being of the precise form and arrangement of those which furnished steam for the Centennial engine. The aggregate quantities of the three boilers are as follows:

Water-heating surface	1231 square feet.
Superheating surface	508 " "
Grate surface	45 " "
Ratio of heating to grate surface	38.6 to 1.

The original contract contained a clause that when the engine was completed and ready for service two tests should be made: one to continue 24 hours incessantly, and the other to occupy two weeks, omitting the intervening Sunday, running the fires ten hours per day as nearly as may be, with the engine at such speed as shall guarantee the delivery of 3,000,000 gallons in 24 hours. In this every day running test, all the fuel shall be taken into account, the wood used for kindling being reckoned at 45 per cent. of its weight in coal. No allowance to be made for ashes or clinkers. The coal used in these tests to be first-class run of the mine. The quantity of water to be delivered is to be ascertained by weir

measurement. A duty of 100,000,000 foot-pounds was guaranteed on the 24 hours' trial, and 80,000,000 foot-pounds on the two weeks' trial.

The two weeks' every day running test was begun August 6, 1878, and the computation of the duty for the average of the twelve days is as follows:

Number of hours run	120	
" revolutions	373,124	
Average reading of water-gauge, in pounds	109.164	
" " draft-indicator, being distance from centre of water-gauge down to surface of water in pump-well, in feet	16.3169	
" of total head against pump, in feet	269.0185	
" temperature of water, degrees Fahrenheit	78.194	
Weight of a cubic foot in pounds	62.20625	
Displacement of one revolution in cubic feet	5.6952	
Wood used to start fire, in pounds, 2126.5 =	956.925	coal.
Coal used to start fire, in pounds	6521	
" while pumping, in pounds	26,598.5	
Total pounds of fuel used	34,076.425	

DUTY.

$$\frac{5.6952 \text{ cubic feet} \times 373{,}124 \text{ revolutions} \times 62.20625 \text{ pounds} \times 269.0185 \text{ feet} \times 100}{34{,}076.425 \text{ pounds of coal}}$$

= 104,357,654 foot-pounds per 100 pounds of coal.

The weir measurement was made for four hours on August 28, and showed the capacity of the engine at the average speed of the two weeks' test to be 3,060,000 gallons in 24 hours.

The twenty-four hours' test was begun on October 3 following; the details and computations of the duty are as follows:

Number of revolutions	74,463
Average reading of water-gauge, in pounds	108.06
" " draft-indicator, in feet	16.6706
" of total head against pump, in feet	266.4852
" temperature of water	68.458°
Weight of a cubic foot, in pounds	62.2882
Displacement of one revolution, in cubic feet	5.6952
Coal used while pumping, in pounds	5272

DUTY.

$$\frac{5.6952 \text{ cubic feet} \times 74{,}463 \text{ revolutions} \times 62.2882 \text{ pounds} \times 266.4852 \text{ feet} \times 100}{5272 \text{ pounds of coal}}$$

= 133,522,060 foot-pounds per 100 pounds of coal.

The results show that the engine exceeded the guarantee on the two weeks' every day running test by more than 30 per cent., and on that of the 24 hours' test by more than 33 per cent.

Annual Duty.—During the fiscal year ending November 30, 1888, the water-works report claimed an annual duty of 123,656,000 foot-pounds, largely exceeding its previous records. As no determination of the steam-consumption and boiler-evaporation had ever been published, the determinations of duty depending on measurements of coal and water-pressure only, Professor J. E. Denton requested permission to make these measurements and to verify the extraordinary duty contained in the above report. This permission was cordially granted, the management co-operating with Professor Denton in every possible way to make the investigation complete.

Professor J. E. Denton's experiments had for their special object the measurement of the steam consumed by the engine per horse-power under various conditions calculated to afford scientific data regarding the application of thermodynamic principles in engines of this type. This required the separation of the performance of the boiler from that of the engine. The engine had been several times previously tested without making this separation, the performance being stated in foot-pounds of "duty," which expresses the combined economy of the engine and boiler.

PROGRAMME OF EXPERIMENTS EXECUTED.

1. From May 30 to June 1, 1889, inclusive, the engine was run continuously 72 hours under its usual working conditions, and measurements made of

a. The coal consumed, anthracite, stove-size, being used.

b. Horse-power performed both for pumping the water and to overcome the friction of the engine itself.

c. The pounds of steam generated in the boilers and used to operate the engine.

d. The pounds of water used to condense the steam as it left the engine.

e. All temperatures and detailed measurements necessary to determine the heat rejected by the engine, the steam condensed in the jackets, the economy due to passing the condensed steam from the receiver through heater in chimney, and the dryness of the steam.

f. The duty of the engine, the steam per hour per horse-power, and the evaporation of the boilers per pound of coal, were the principal quantities sought.

2. During June 1 and 2 the engine was run 24.13 hours under the same conditions as before, but with selected George's Creek bituminous coal for fuel.

The same determinations were made, but the special object was to check the previous test and at the same time ascertain how much gain in duty could be expected from choice bituminous coal as compared to the stove anthracite used in the first test, and in the usual operation of the plant.

3. During June 3, 4, and 5, short tests of from six to nine hours were made, the engine being run with and without steam in its steam-jackets. No measurement of fuel was made, the object sought being to determine the steam consumed by the engine per horse-power with and without the jackets in action.

General Arrangement of Apparatus.—In Fig. 256 is shown a sectional view of the engine with the various thermometers and meters attached. Thermometer *G* measures the temperature of the feed-water as it enters the boiler. Thermometer *M* was a special one 36 inches long, graduated by H. J. Green. Twenty inches of length were devoted to the 100 degrees between 300° and 400° Fahr. Each degree was therefore represented by a fifth of an inch, thus making the temperature of the steam measurable with great accuracy. This thermometer remained at about 356.1 degrees during most of the experiments, thereby showing the steam to be superheated about 3 degrees.

For a short time the chimney temperature rose from 428 degrees, its normal amount, to 500 degrees, and the super-

heating then became equal to 18 degrees, but the average was 3 degrees, and the minimum 2½ degrees.

The quality of the steam was therefore determined to be perfectly dry on entering the engine without the application of the more or less vexatious devices known as calorimeters, for the use of one of which, the Barrus superheating form, arrangements had been made.

Thermometer *J* determined the temperature of the water from the suction-well on its way into the condenser.

Thermometers *H* and *I* measured the temperature of the water leaving the condenser, *H* extended just through the pipe and *I* to nearly the opposite side of the pipe; the former showed constantly about 104 degrees when the latter read about 92 degrees. This apparent anomaly was finally explained by the conclusion that the thermometer *H* showed the temperature of saturation of the steam or vapor at the pressure given by the vacuum-gauge, 28½ inches, while *I* gave the temperature of the mixture of condensing water and condensed steam.

Thermometer *N* showed the temperature of the hot-well water. Its readings were practically identical with those of *I*, and practically with those of *G* when the latter were not influenced by the entrance of the condensed steam from the jackets into the feed-pipe. Such steam was pumped into the feed-pipe at a point just below *M*.

I. Principal Results—Anthracite Test.—Date, May 30 to June 1, 1889. Duration, 72 hours.

Revolutions per minute	49.03
Boilers in use	3
Pressures:	
Average boiler-pressure	127 pounds.
" receiver-pressure	8⅓ "
" vacuum	28¼ inches.
" back-pressure in low-pressure cylinder	1.25 pounds.
" water-pressure against pumps	115 "
" suction against pumps	5.78
" barometer-pressure	30

Temperatures:		
Temperature of saturated steam at average boiler-pressure	353	deg. F.
Average temperature of steam in steam-pipe two feet from steam-chest	356	"
Degrees of superheating	3	"
Temperature of feed-water entering boilers	104	"
" " leaving hot well	92.1	"
Increase of temperature due to admission of jacket-water	11.9	"
Temperature of jacket-water entering feed-pipe	310	"
" condensing water leaving suction-well	63	"
" gases at exit from boiler-flues	428	"
" gases after passing heater in flue through which condensation from receiver was circulated	354	"
" water blown to waste from receiver	225	"
" the atmosphere	70	"
Total quantities:		
Anthracite coal	15,710	pounds.
Equivalent of wood at 40 per cent., 263 pounds	105	"
Total	15,815	"
Ashes, clinkers, etc.	2,238	"
Water fed to boilers, including condensation from jackets	140,969	"
Condensing water	4,201,337	"
Hourly quantities:		
Fuel	219.5	"
Feed-water	1,958	"
Condensing water	58,351	"
Condensed in jackets	107	"
" through receiver	156	"
Blown to waste from receiver	118	"
Average indicated horse-power, steam-cylinders	143.89	
" " " of pumps	138.36	
" horse-power to overcome friction of engine, including air-pump	5.13	
" horse-power to operate air-pump	2.35	
Rate of combustion, coal per square foot of grate per hour	4.9	"
Economy of engine:		
Steam per hour per indicated horse-power of steam-cylinders	13.64	
This figure should be regarded subject to a possible error of 3 per cent. due to this limit of error affecting the determination of horse-power from indicator cards.		
Efficiency of mechanism of engine and pumps	96.4	per cent.

Economy of boilers:	
Water evaporated per pound of fuel from 102 degrees into steam at 125 pounds pressure . . .	8.88 pounds.
Equivalent evaporation from and at 212 degrees per pound of combustible	12.12 "
Economy of engine and boilers combined:	
Duty per 100 pounds of coal at actual efficiency of boilers	124,720,000 foot-pounds.
Coal per hour per indicated horse-power of steam-cylinders	1.54 pounds.
If boilers had evaporated 10 pounds of water into steam from actual temperature of feed-water, the duty would be per 100 pounds of coal	140,430,000 foot-pounds.
Coal per hour per indicated horse-power of steam-cylinders	1.37 pounds.

CHECK UPON MEASUREMENT OF FEED-WATER BY MEASUREMENT OF HEAT EXHAUSTED OR REJECTED AT AIR-PUMP OF ENGINE.

The heat which entered the engine per hour was as follows: 1958 pounds of feed-water, to each pound of which there was supplied 1221 heat-units, the total heat of evaporation, hence:

Heat given to steam $1958 \times 1221 =$	2,390,718 heat-units.	
The chimney-heater evaporates 38 pounds of water at 225 degrees in steam at 22 pounds pressure, making $38 \times 960 =$	36,480	
Total heat received by engine		2,427,198

This heat is distributed as follows:

58,351 pounds of condensing water is raised in temperature from 63° to 92.1° Fahr., making	1,698,014	
118 pounds of the feed-water is blown to waste from the receiver, representing $118 \times 225 =$	26,550	
107 pounds of the feed-water is drained from the jackets, representing $107 \times 310 =$	33,170	
The remainder of the feed-water, or $1958 - 118 - 107 =$ 1733 pounds, is found in the hot well, representing $1733 \times 92.1 =$	159,609	
The indicated work performed, or 143.49 horse-power, represents $\frac{143.49 \times 1,980,000}{772} =$	368,769	
The estimated radiation is 20 pounds of steam per hour . .	38,320	
		2,324,432
Balance unaccounted for		102,766

This discrepancy represents 4 per cent. of the whole amount of heat. Part of this is ascribable to the leakage of steam from the safety-valves and stuffing-boxes. The remainder is due to undefined causes. The error is, however, on the right side to make the feed-water measurements worthy of confidence as being slightly too large and unfavorable to an exaggeration of the economy of the engine.

II. Principal Results—Bituminous Test.—All data for this test are the same, except the following:

Revolutions per minute	49.44
Average boiler-pressure	128 pounds.
" water-pressure against pumps	116 "
" cut-off high-pressure cylinders	0.235
" horse-power indicated, steam-cylinders	147.95
" " " pumps	140.74
" " " devoted to friction	7.21
Total bituminous coal	5200 "
Total wood = 40 per cent. × 200 =	80 "
Total	5280 "
Total ashes	510 "
Duration of test, 24 hours, 13 minutes.	
Fuel per hour	218.6 "
Feed-water per hour	2046 "
Economy of engine:	
Steam per hour per indicated horse-power of steam-cylinders	13.82 "
Efficiency of mechanism	95 per ct.
Economy of boilers:	
Water evaporated per pound of fuel from 102 degrees into steam at 126 pounds	9.35 pounds.
Water evaporated per pound of combustible, and at 212 degrees	12.11 "
Economy of engine and boilers combined:	
Duty per 100 pounds of coal at actual efficiency of boiler	127,350,000 ft. lbs.
Coal per hour per indicated horse-power of steam-cylinders	1.48 pounds.
Economy of engine and boilers combined if boilers had evaporated 10 pounds of water per pound of coal from temperature of 102 degrees:	
The duty per 100 pounds of coal =	145,041,000 ft. lbs.
Coal per hour per indicated horse-power of cylinders	1.30 pounds.

By reference to the theoretical heating capacity of the two kinds of coal it will be seen that the increase of duty with the bituminous coal was not as great as chemical analysis indicates should have been the case.

Tests to determine Influence of the Steam-Jackets.—The results of these tests indicate that the saving by the use of the steam in the jackets is not greater than three per cent. of the fuel used without steam in the jackets.

Economic Value of the Flue Heater.—Assuming the effect of the heater to be the addition of about 40 pounds of steam to the volume of the latter, which remains vaporous at the end of admission to the low-pressure cylinder, the useful effect would be that due to increasing the admission-pressure of the low-pressure card. In round numbers, the steam remaining vaporous at low cut-off represents 1600 pounds per hour. The addition of 40 pounds would increase the pressure $\frac{40}{1600} = \frac{1}{40}$. The absolute pressure averages 22 pounds. Hence the increase of average pressure during admission would be less than $\frac{6}{10}$ pound per square inch. The gain in mean effective pressure due to such an increment would represent about one per cent. of the total power of the engine.

Reynolds's Triple-Expansion Pumping Engine.—The engine illustrated in the folding-plate, Fig. 258, is one of five built by the Edward P. Allis Company, of Milwaukee, Wis., for the Chicago Water-Works, and is of the same type and general design as engines built for the water-works of Milwaukee, Omaha, Denver, Detroit, New Orleans, and several other cities.

The original engine of this type was built in 1886 for the high service of the Milwaukee Water-Works, and was the first triple-expansion engine built for water-works purposes. This original engine, although operated with only 80 pounds steam-pressure, has given a duty as high as 128,000,000 foot-pounds

for each 100 pounds of anthracite coal burned, and has developed an indicated horse-power on 1.35 pounds of coal per hour.

The following description of the Chicago engine covers the main features of all the triple-expansion pumping engines built by this firm. The general arrangement of the engine above the floor is somewhat similar to the marine type of engine. The high-pressure cylinder is 27 inches diameter by 60 inches stroke, the intermediate cylinder 46 inches diameter by 60 inches stroke, and the low-pressure cylinder 70 inches diameter by 60 inches stroke. The steam-cylinders are steam-jacketed throughout and are fitted with Reynolds-Corliss valve-gear, the valves being placed in the cylinder-heads; this arrangement reducing the clearances to the very smallest amount, in some cases the total clearance being only three-fourths of one per cent. of the cylinder volume.

The valve-gear of each cylinder has adjustable trip cut-offs, the high-pressure valve-gear being also under the control of a speed-governor. Each cylinder has two piston-rods connected to a forged steel cross-head, from which four rods extend directly to the heads of the pump-plungers, which are situated below and directly in line with the steam-cylinders. This arrangement makes the machine direct-acting in the best sense of the word, as the power is transmitted directly from the steam-pistons to the pump-plungers, without passing through links, beams, or swinging journals of any kind, the only power passing through the crank-shaft being that which is transmitted to the fly-wheels at the beginning of the stroke, and returned from them near the end of the stroke.

There are three cranks set at angles of 120 degrees apart, and each crank being connected to its particular plunger insures a practically uniform flow of water from the pumps, and reduces the shock and jar on the pumps and pipes to a minimum.

The main shaft is 16 inches diameter in the journals and 18 inches in the wheels, the main journals being 20 inches long. The fly-wheels are two in number, and weigh about 25

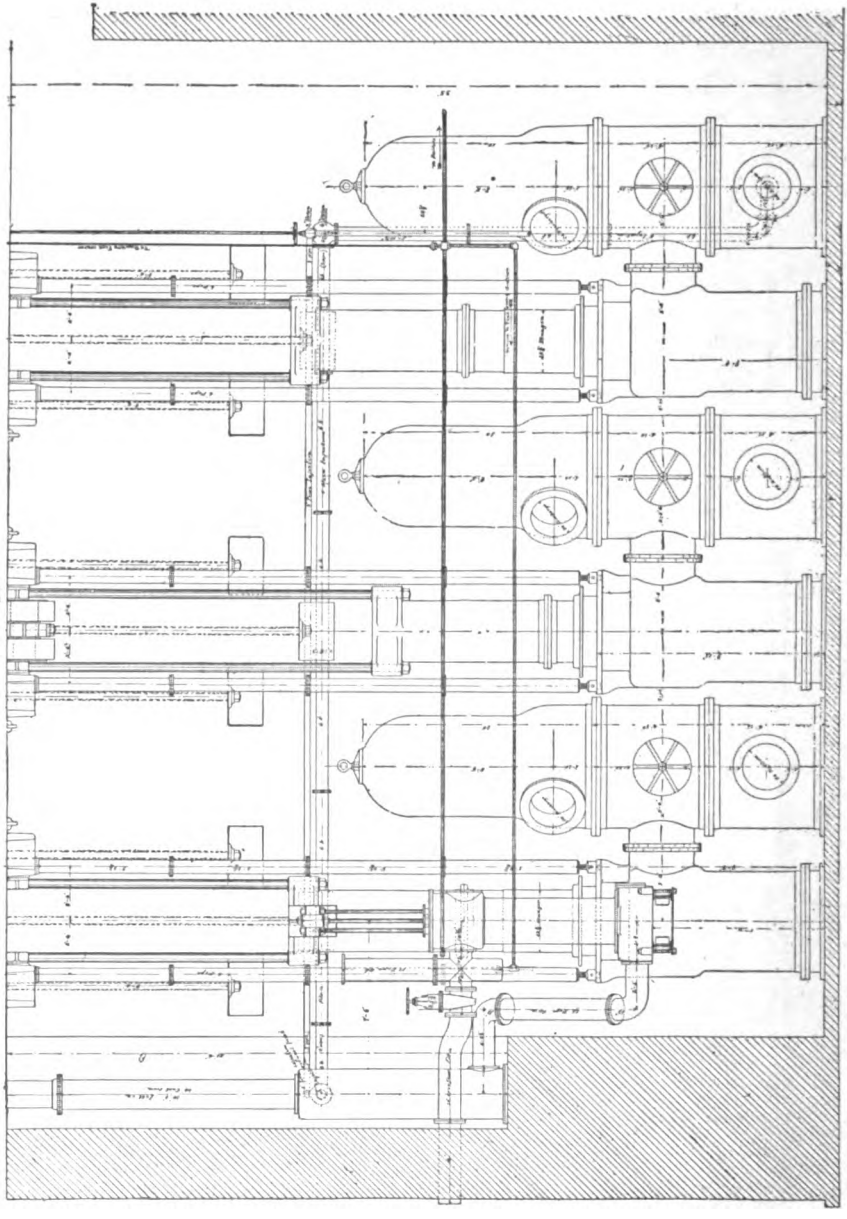
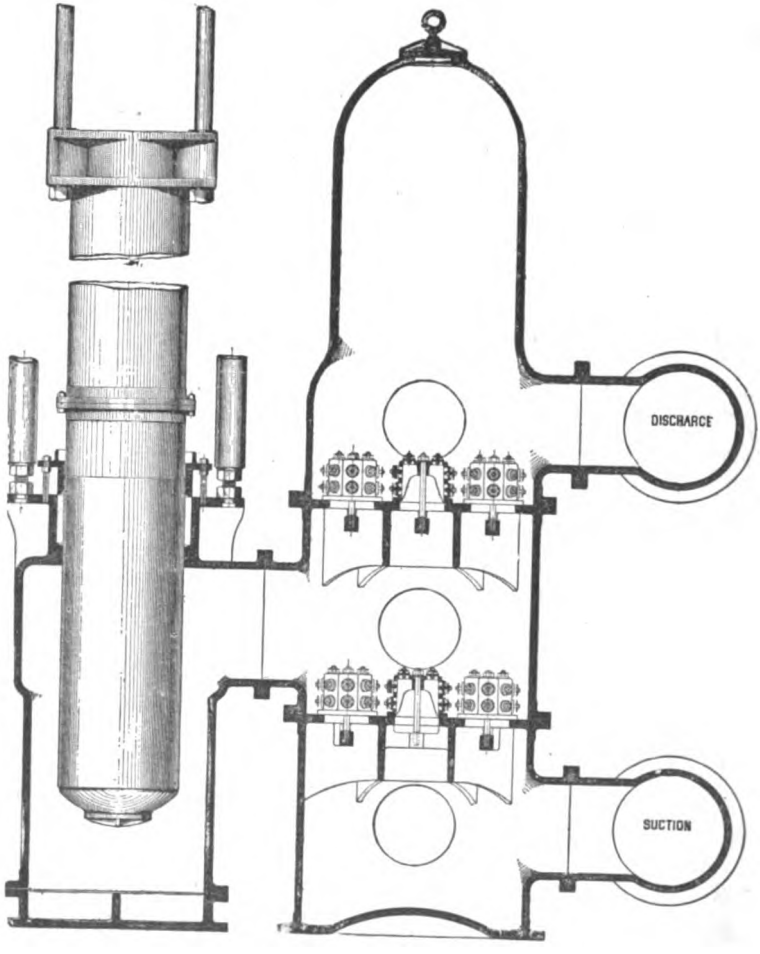


FIG. 259.

tons each. The pumps are three in number, being placed directly under the steam-cylinders, and arranged as shown in Fig. 259. They are of the single-acting, outside-packed type,

FIG 260.

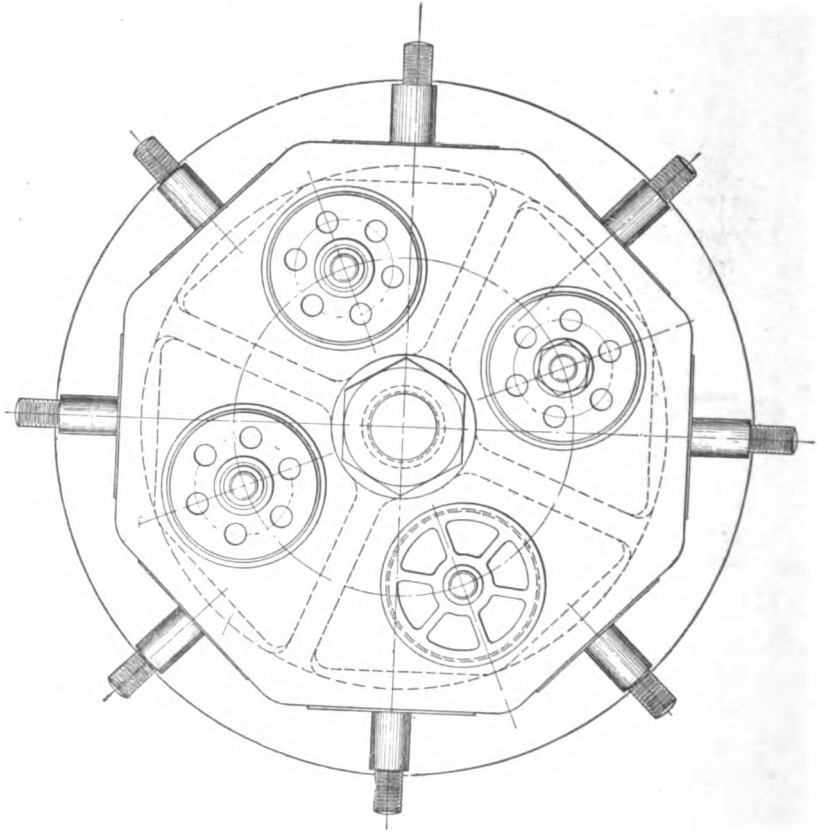


as shown in Fig. 260, having plungers 32¾ inches diameter by 60 inches stroke, and the plungers are so counterbalanced

that the work done in the steam-cylinders is the same on the up-stroke that it is on the down-stroke.

The pump-barrel and valve-chambers, shown in sectional elevation in Fig. 260, are extremely simple in construction.

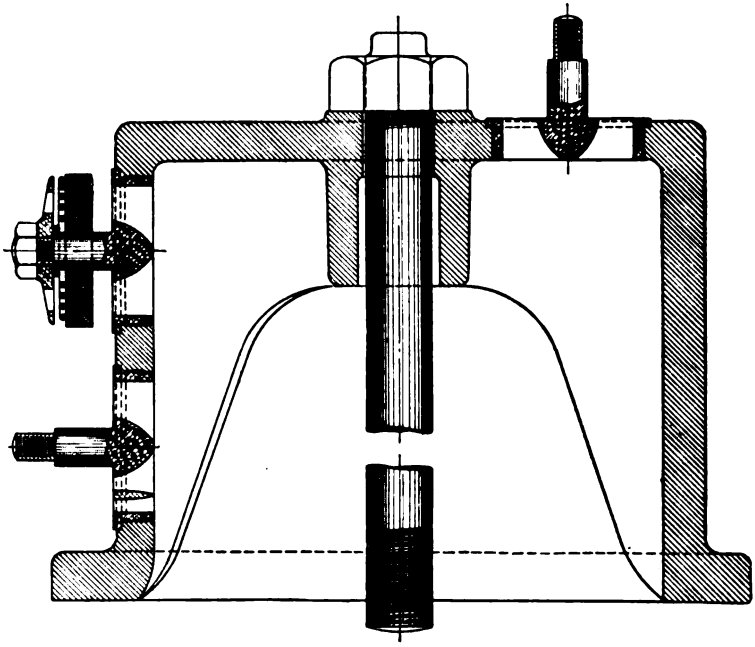
FIG. 261.



The valves used are of rubber, of small diameter, working on brass seats; a section through a valve and seat is shown in Fig. 262. The valves are arranged in groups mounted on cages; a plan of one of these cages in shown in Fig. 261, and in sectional elevation in Fig. 262, each cage being re-

movable independent of the others, or permitting the removal of a single valve without disturbing a cage. These cages are held in place by a single bronze tap-bolt extending through

FIG. 262.



the centre of the cage and screwing into a bridge cast in the valve-chamber, as shown in Figs. 260 and 262.

The air- and feed-pumps are driven by an arm attached to the low-pressure plunger, thus giving them the same stroke as the main pumps. The height of the engine above the main floor is about 35 feet, and the depth of pump-pit is 35 feet, making a total height of 70 feet.

The weight of one engine complete with its pumps is 450 tons. At Chicago two of these engines are in operation at the West Harrison Street Station and three in the new station at Fourteenth Street. The Harrison Street engines have been in operation about two years and a half; the Fourteenth Street

engines have recently been put in operation on the completion of the new four-and-a-half-mile intake tunnel. These engines have a nominal capacity of 15,000,000 gallons each, against a head of 125 feet, when running at a speed of about 16 revolutions per minute, and have a guaranteed capacity of 18,000,000 gallons in 24 hours under the same head.

Duty-Trial.—A test of the Harrison Street engines made by Mr. Bernard Fiend, assistant engineer of Chicago, in October, 1891, showed the following results:

Duration of test, from 8.45 A.M. to 4.45 P.M.	8 hours.
Total revolutions of No. 1 engine	7,972
" " No. 2 "	8,042
" " both engines	16,014
Average " No. 1 engine	16.608 per min.
" " No. 2 "	16.754 "
" " both engines	16.681 "
" steam-pressure	124 86 pounds.
" " in first receiver	25.83 "
" " in second receiver, atmosphere	Zero.
" vacuum in inches	27.0 inches.
Total head of water pumped against	126.65 feet.
Water fed to boilers, 197 barrels, at 400 pounds each plus 163 pounds	78,963 pounds.
Average temperature of feed-water	115 deg. F.
Total coal burned	10,975 pounds.
Total ashes	1,408 "
Per cent. of ash	12.83 per cent.
Actual evaporation from 115°	7.195 pounds.
Evaporation from 212°	8.238 "
" " per pound of combustible	9.45 "
Total number of U. S. gallons pumped by both engines	10,511,348 gallons.
" " pounds of water pumped by both engines at 8.34 pounds per gallon	87,664,640 "
Total net foot-pounds of work done by both engines	11,087,822,390 ft. lbs.
Duty per 1000 pounds of feed-water on plunger displacement	140,416,679 "
Net duty after allowing 2 per cent. for slip and leakage	137,608,346 "
Average indicated horse-power of No. 1 engine	388.5 H. P.
" " " No. 2 engine	390.2 "
" total indicated horse-power of both engines	778.7 "
Water per hour per indicated horse-power	12.675 pounds.

Reference to the indicator diagrams taken during the test, average samples of which are reproduced on smaller scale in

FIG. 263.

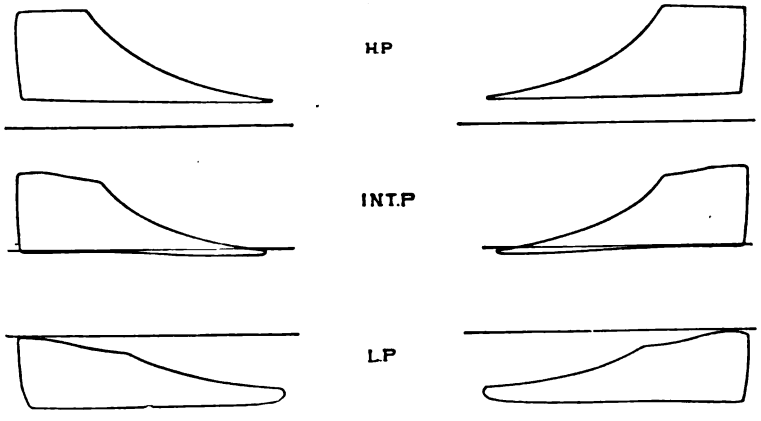
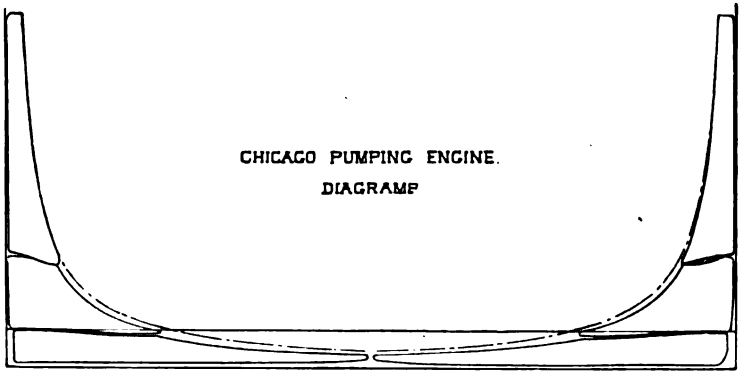


Fig. 263, together with the combined diagram Fig. 264, will show how close the actual expansion of steam conformed

FIG. 264.



to the theoretical, and probably no better results have ever been attained under similar conditions.

As an illustration of the economy of this type of engine in every-day operation under ordinary conditions of service, it is

reported that the 18,000,000-gallon engine of this type in the Milwaukee Works, already referred to, attains a monthly duty of about 120,000,000 foot-pounds for each 100 pounds of coal burned for all purposes without deductions of any kind, the coal burned being only a fair quality of anthracite, and the boilers being of the ordinary horizontal tubular type, evaporating about 8¾ pounds of water per pound of coal.

INDEX.

A.

Accumulator, 162.
friction in, 165.
Acid water, effect of, 290.
Air, clearing pumps of, 112.
quantity to resist shock, 118.
Air- and vacuum-chambers, 90.
Air-chamber, fire-pump, 282.
lined with wood, 310.
mine-pump, 295.
pressure-pump, 164.
Air-pump, bucket and piston, 26.
for air-chambers, 93.
for condenser, 27.
American Society of Mechanical Engineers, 71, 321, 334, 363.
Andrews's centrifugal pump, 357.
Annular valve, 64.
Appold's centrifugal pump, 335.
Area of steam-ports, 237.
for valve-seats, 84.
Artesian-well pump, 223.
Assoc. Fac. Mut. Ins. Co., 271–288.
Atmospheric pressure, 9.
Atmospheric pump, 10.
Automatic relief for pressure-pumps, 169
sprinklers, 274.
Auxiliary throttle-valves, 316.

B.

Ball-valve, 55.
with metal core, 56.
Barrus, George H., 346, 412.
Bell-valve, 55.
Berrenberg rotary pump, 226.
Bessemer works, pumps for, 167.
Birkenbine, H. P. M., quoted, 175.
Björling, P. R., quoted, 53.
Blake valve motion, 204.
Boiler performance, 363.
test by Prof. Unwin, 382.
tests and duty-trials, 362.
form recommended, 363.
Boilers, Pawtucket Water-Works, 422.
heat used and lost in, 384.
Lynn Water-Works, 417.
Brass-covered piston-rods, 277.
Brewery-pump, 145.
Brick-earth, pumping, 189.
Bucket- and piston-pump, 33.
Bucket- and plunger-pump, 32.
Bucket-piston for air-pump, 26.
Bucket-pump, ball-valve, 25.
mitre-valve, 25.
Bucket-pumps, 23.
Buffalo Stm.-Pump Co., Mine P'ps, 304.
Built-up wing-valve, 67.
Bushings for valve-seats, 82.
Bushing stuffing-boxes, 45.
for plunger, example, 309.
Butterfly-valve, 51.
By-pass connection, 145.

C.

Calorimetric value of coal, 383.
Cameron, A. S., 387.
Cameron's high-duty engine, 387.
sinking-pump, 291.
valve motion, 203.
valve-seats, 61.
Cataract for steam-pumps, 220.
Centrally-packed plunger-pump, 31, 142.
Centrifugal force in pumps, 338.
pumps, 334.
and engine, 354.
blades for, 336.
casing for, 337.

Centrifugal pumps, concave arm wing, 351.
design, 337.
double suction, 253.
dredging, 354.
effect of high rotation, 340.
hollow arm piston, 351.
in series, 360.
piston for sand, 352, 354.
pistons, forms of, 351.
speed, Morris Machine Works, 356.
whirling in, 336.
Charging-pipe, 122.
Check-valve, 122.
Chicago Water-Works, Reynolds's pumping-engine, 430.
arrangement of pumps, 432.
duty-trial, 436.
section of pump, 433.
valves and cages, 434.
Circular steam-valves, 253.
Clack-valves, 47.
and supplemental valve, 49.
experience with, 48.
fulcrum for, 52.
hinged, 51.
lift of, 52.
Clarkson valve motion, 207.
Classification of pumps, 11.
Clearance, steam-cylinder, 280, 407.
Coal, calorimetric value of, 383.
Cold-rolled steel, 37.
Cold-water valves, 56.
Coles, H. J., quoted, 119.
Collars for plunger-rods, 43.
Combined foot-valve and strainer, 104.
Compensators. (See High-Duty Attachment.)
Compound cylinders, ordinary sizes, 250.
ratios, 249.
stuffing-box, 245.
engine piston-rods, 240.
power of, 256.
isochronal pump, 262.
mine-pumps, 314.
Compound mine-pumps, Buffalo, 304.
Jeanesville, 313.
Knowles, 305.
pressure-pump, 168.
pump, Tangyes's, 259.
pumps, duty-trial, duplex, 370.
fly-wheel, 372.
duplex performance, 251.
high-service attachment, 237.
intermediate head, 238.
single, direct-acting pump, 224.
steam-pumps, 235.
action of steam in, 245.
arrangement of, 240.
tank-engine, 254.
Concentric ring-valve, 87.
Conical mitre-valve, 53.
springs, 60.
Corliss, George H., 269, 413.
pumping-engine, Pawtucket, 252, 418.
Cornish double-beat valves, 70–77.
pumping-engine, 310.
clearance in, 407.
Cost of dredging with centrifugal pump, 359.
of fire-pumps, 287.
of pumping earth, 190.
Cranes, hydraulic power for, 168.
pressures in, 152.
Crank, advantages claimed for, 175.
Crank-pumps, 173.
details, 182.
diagram of effort, 176.
geared, 197.
Goodbrand & Co., 185.
Guild & Garrison, 184.
single, 190.
slotted cross-head, 184.
strength for, 174.
valve areas for, 175.
without fly-wheel, 178.
Woodward's, 183.
Cross-head for compound engine, 243.
Cup-leathers, mould for making, 17.
packed piston, 16.

'Cushion-valves, 280.
Cylinders, arrangement of comp., 240.
lagging of, 253.

D.

Dash relief-valve, 228, 237, 280.
Davey, Henry, 405.
high-duty engine, 405.
Davidson triple-expansion engines, 264.
valve-motion, 212.
valve-seats, 61.
Davies, J. D., 390.
high-duty engine, 390.
Davis, E. F. C., quoted, 321.
Dean Brothers' valve motion, 209.
Deane's sinking-pump, 291.
Deep-well pump, 223.
Delivery-pipes, size of, 107.
Denton, Prof. J. E., 39.
on steam-jackets, 253.
Differential plunger-pump, 143.
Dimensions, Corliss engine, Pawtucket, 421.
Leavitt engine, Lynn, 416.
Reynolds's engine, Chicago, 431.
Direct-acting mine pump, 295.
steam-pumps, 200.
Disk-valves, 56.
Double-beat valves, Cornish, 70.
India-rubber valves, 70.
Double-plunger pressure-pump, 169.
Drainage of steam-jackets, 251.
Drainage-pipe, 151.
Dredging by centrifugal pumps, 355.
General Gillmore on, 357.
Drip-cocks, 284.
Duplex pumps, 226.
fire-pumps, 272.
mine-pumps, 297.
sinking-pump, 294.
steam-valves, 229.
tanking, 254.
valve-adjustment, 230.
valve-gear, 227.
valve-movement, 233.
Duty-trials of pumping engines, 361.
Duty-trials of pumping engines, compound duplex, 370.
fly-wheel, 372.
duration of test, 375.
engine performance, 367.
English method, 376.
guaranteed performance, 374.
Lynn pumping engine, 418.
measuring work done, 366.
Milwaukee pumping engine, 438.
Pawtucket pumping engine, 423–429.
plunger displacement, 366.
quantity of work done, 366.
standard proposed, 363.
Worthington engine, 385.

E.

Easy-seating water-valve, 63.
Eccentric, Woodward pump, 184.
Eccentrics for pumps, 195.
Efficiency, Appold's pump, 340.
centrifugal pump, 337.
curves, Webber, 343.
Gwinne pump, 340.
large centrifugal pumps, 349.
loss of, in pumps, 149.
rotary pumps, 331.
24-inch centrifugal pump, 346.
Electrical pump, 198.
Ellis, G. A., quoted, 108.
Engine, efficiency, 412.
Engineering News, quoted, 256, 264.
English method duty-trials, 376.
Erste Brünner Masch. Fab. Pump, 136.
Evaporation assumed, duty-trials, 362.
Expansion diagrams, 179.
Expansive use of steam, 177.

F.

Factor of safety, 277.
piston-rods, 38.
pressure-pumps, 163.
Farcot's centrifugal pumps, 350.
Fastening piston-rods, 40.
Fielding, John, high-duty engine, 408.
Fiend, Bernard, duty-test by, 436.

Final results of boiler-trial, 363.
Fire-pumps, 271.
air-chamber, 282.
brass fitting of, 277.
capacity of, 274.
commercial sizes, 275.
cost, 287.
duplex, 272.
force-valves, 278.
H. P. required for, 273.
hose connections, 285.
necessary fittings, 287.
pipe sizes, 281.
piston pattern, 276.
speed, 99.
plunger pattern, 276.
pressure-gauge, 283.
priming-pipes, 284.
ratio of steam to water, 276.
safety-valve, 283.
size-plate, 276.
sizes recommended, 273.
speed of, 274.
steam-ports, 279.
strength of parts, 277.
suction-valves, 278.
test for acceptance, 285.
vacuum-chamber, 281.
Fire-stream, size and capacity, 274.
Five ports in duplex cylinders, 226.
Flanging machines, pressures in, 152.
Flow of water into a pump, 96, 114.
Floyd and Morton's steam-end, 259.
Fly-wheel, function of, 178.
pumping engines, 411.
Foot-valve and strainer, 103.
multiple, 105.
relief-valve, 104.
single disk, 105.
Force of impact in pumps, 339.
Form of air-chamber, 91.
Foundry-lifts, pressures in, 152.
Four-seated valve, 80.
Freeman, John R., 287.
Friction in hydraulic machines, 165.
loss, table of, 108.
Frictional resistance of water, 150.
Fulcrum for clack-valves, 52.

G.

Gain by compounding, 235.
Geared crank-pump, 197.
pressure-pump, 167.
pump with eccentrics, 195.
Gillmore, General Q. A , quoted, 357.
Goodbrand & Co. crank-pump, 185.
Gordon isochronal compound pump, 262.
valve motion, 218.
Green, Howell, quoted, 310.
Greindl rotary pump, 329.
Grooved water-piston, 19.
Groshon, John A , 400.
high-duty engine, 400.
Guaranteed performance of pumping engine, 374.
Guild and Garrison crank-pump, 184.
valve motion, 215.
Gun-metal-lined mine-pump, 308.
Gwinne pump, efficiency of, 340.
tests, 350.

H.

Hale, R. A., quoted, 331.
Heat-units, a base for duty-trials, 363.
Heat used and lost in boilers, 384.
Hemispherical valve-guard, 60.
Henthorn, J. I., quoted, 411.
High-duty attachment, Cameron's device, 387.
Davey's device, 405.
Davies's device, 390.
Fielding's device, 408.
Groshon's device, 400.
Worthington's device, 393.
High-service attachment, 237.
Hill, John W., quoted, 256.
Hinged clack-valve, 51.
Hippopotamus leather, 48.
Holly, A. L., quoted, 167.
Horizontal strainer, 102.
Hot water, pumping, 119.

Hot-water valves, 56.
Hydraulic elevator service, 13.
engines, 165.
pressure-pumps, 152.
pressures, notes on, 153.
pump, water-end, 155.
transmission of power, 164.
Hyperbolic logarithms, 180.

I.

India-rubber disk-valves, 56.
Indicator diagram, pump, 50.
Inside-packed plunger-pump, 29, 138.
Intermediate cylinder-head, 238.
head with stuffing-box, 245.
Irrigating machinery, Pacific coast, 360.
Isochronal pumping engine, 262.

J.

Jamb-nuts, 44.
Jeanesville mine-pump, 313.
Joints for mine-pipes, 324.

K.

Knowles compound mine pump, 305.
piston mine-pump, 296.
valve motion, 201.

L.

Lagging steam-cylinders, 253.
Lawrence, Mass., pumping engine, 269.
Mach. Co. centrifugal pump, 344.
Leaking suction-pipes, 100.
Leather-faced wing-valve, 53.
Leather for packings, 18.
Leathers for clack-valves, 47.
Leavitt, E. D., Jr., 269, 413.
compound pumping engine, 413.
Length of pump-stroke, 126.
Lift of butterfly-valves, 52.
of clack-valves, 52.
Limit of suction, 10.
Lining piston-pumps, 20.
mine-pumps, gun-metal, 298, 308.
wood, 308.
removable, 22.
Loss of efficiency in pumps, 149.
Lost motion in duplex pumps, 229.
Lost-motion links, 230.
Lubricating compound mine-pumps, 296.
Lynn, Mass., pumping engine, 413.

M.

McCarty's centrifugal pump, 335.
McCreath, James, quoted, 110, *et seq.*
Machines for raising water, 7.
Mair, John G., quoted, 176.
Marsh valve motion, 213.
water-valve, 63.
Mean pressure of steam, 181.
Mechanically-operated valves, 88.
Metal disk-valve, 58.
Metallic packings for pump-rods, 46.
Metal rings for water-pistons, 18.
used for mine-pumps, 289.
Milwaukee pumping-engine duty, 438.
Mine-pumps, 289.
arrangement of, 307.
Buffalo S. P. Co., 304.
centrally packed, 300, 302.
compound, 314.
duplex, 297.
electrical, 199.
end-plungers, 303.
lined, gun-metal, 298, 308.
wood, 308.
lubricating piston of, 296.
multiple valves for, 302.
piston-pattern, 299.
plungers, 318.
pot-valve chambers, 306.
rods, 300.
rotative, 408.
single, 295.
valve-area, 302.
valve-chambers, 309, 321.
valve-seats, 62, 300.
valves, 300, 319.
water-ends, 311.
Mine-water, action on metals, 319.
effects of, 290.
pipes, 324.

Mitre-valve, conical, 53.
Montreal sewage-pumps, 346.
Morris Machine-Works, 351.
Multiple-beat valve, 68.

N.

Nagle, A. F., quoted, 71.
Noisy pumps, 87.
Non-conductors for lagging, 253.
Nozzle, size for fire-pump, 285.

O.

Outside-packed plunger-pump, 30, 139.

P.

Packed plunger-pump, 135.
Packing a water-piston, 16.
Parallel spiral springs, 60.
Parsons, R. C., quoted, 334, *et seq.*
Passages through centrifugal pumps, 337.
Pawtucket pumping engine, 269.
annual duty, 423.
Denton's experiments, 424.
flue-heater, 430.
general arrangement, 425.
performance of boilers, 363.
of engine, 251.
results, anthracite, 426.
bituminous, 429.
steam-jacket, 430.
Perreaux's valve, 81.
Pipe sizes, fire-pumps, 281.
Piping a pump, 121.
Piston water-end, removable lining, 130.
Piston- and plunger-pump, 34.
rods, 37.
Piston-pump, ball-valves, 129.
Erste Brünner, 136.
for mine, 298.
for thick stuff, 145.
linings, 20.
with wing-valves, 128, 145.
Piston-rods, brass-covered, 277.
compound engines, 240.
fastenings, 40.
Piston-rods, protected, 245.
Piston-speed for pumps, 126.
underwriter pump, 99.
Pistons, cup-leathers, 16.
fibrous packing, 15.
fitted with wood, 19.
lubrication of, 296.
metal rings, 18.
movement with crank, 174.
with grooves, 19.
Plunger and ring water-end, 137.
and solid ring, 28.
bushing, example of, 309.
for mine-pumps, 300, 321.
Plunger-pump, 27.
centrally-packed, 31, 142.
central diaphragm, 30.
differential, 143.
inside-packed, 29, 138.
outside-packed, 139.
outside tie-rods, 30.
vertical, 144.
water-end, 129.
Worthington, 132.
Plunger-rod fastenings, 40.
Poillon, M., quoted, 328.
Pot-valve chamber, 158.
mine-pump, 305.
number of, 161.
Power for hydraulic engines, 165.
hydraulic transmission of, 164.
of compound pumping-engine, 256.
pressure-pumps, 169.
required for pumping earth, 189.
Power-pump, 195.
Pressure affected by accumulator, 163.
caused by shocks, 116.
Pressure-gauge, fire-pump, 283.
Pressure-pumps, 152.
air-chambers for, 164.
automatic relief, 169-171.
compound, 168.
design, 158.
double plunger, 169.
duplex, 167.

Pressure-pumps, factors of safety in, 163.
geared, 167.
large water-ends, 162.
materials for, 154.
power, 169.
valve and seat, 160.
Pressures in hydraulic machines, 152.
of water, table of, 124.
Priming-pipes, 284.
Priming-tank, fire-pump, 284.
Protected piston-rod, 245.
Pump, electrically driven, 198.
speed of, 86.
Pump-bucket for high pressure, 24.
Pump-diagram, vacuum, 50.
Pump-rod collars, 43.
details, 40.
Pump-valve, Corliss's, 421.
Woodward's, 149.
Pumps, clearing of air in, 112.
pressure caused by shocks, 116.
varieties of, 8.
Pumping brick earth, 189.
engines, duty of, 361.
hot water, 119.

R.

Radial force in centrifugal pumps, 339.
Raising water above suction limit, 10.
Ratios of H. P and L. P. cylinders, 249.
Rectangular-valve, double-seating, 67.
Relief-valve and pipe, 122.
for foot-valve, 104.
Removable lining, 22.
Revolutions, duplex, defined, 274.
Reynolds, Edwin, 413.
Reynolds's triple-expansion engine, 430.
Rhinoceros leather, 48.
Richards, John, 360.
Riedler, A., 87.
Riveting, increase of pressure in, 163.
Rood, Vernon H., 313.
Rotary pumps, 325.
classification of, 326.
design, 327.
efficiency, 328–331.
Rotary pumps, failures, causes, 328.
objections to, 326.
Rotative engines for pumping, 408.
Rumsey & Co. rotary pump, 326.

S.

Sand, proportion pumped by centrifugal pumps, 358.
Sand-pumps, 352.
Sewage-pump, centrifugal, 346.
Shand, Mason & Co. pump, 190.
Shears, hydraulic, pressures in, 152.
Shock in pumps, 116.
Side-pipe and strainer, 102.
Simpson, James & Co., 376.
Simpson's lubricating piston, 296.
Single-crank engine, without fly-wheel, 190.
Single mine-pumps, 295.
Sinking-pumps, 290.
Cameron's, 291.
Deane's, 291.
Worthington's. 294.
Size of air-chamber, 92.
for steel piston-rods, 38.
of vacuum-chamber, 96.
of valves, 84.
Slide-valve, duplex, adjustments, 280.
Slide-valves, circular, 253.
Slotted cross head, 184.
Slurry-pump, 189.
Specifications for fire-pump, 271.
Speed and capacity of pumps, 127.
of pumps, 86.
Spindle-valve, 54.
Springs, 58, 278.
Steam, expansion of, 177.
in compound cylinders, 245.
mean pressure of, 181.
per horse-power, 412.
Steam-ends for trade-pumps, 13.
Steam-jackets, 251.
mine-pumps, 316.
value of, 252.
Steam-pipe, 122.
Steam-pipes for collieries, 321.

Steam-ports, area of, 237.
in duplex pumps, 226.
pressure for compounding, 235.
pumps, direct-acting, 200.
underwriter pump, 279.
Steam-valves, circular, 253.
Steel rods, strength of, 37.
Sterk, A. Elink, quoted, 350.
Sticking of valves, 279.
Strainer and vacuum-chamber, 102.
Strainers, 101.
Stroke-gauge, duplex pump, 281.
Stroke-table for pumps, 128.
Stuffing-box bushings, 45.
Stuffing-boxes, 44.
for fire-pumps, 277.
Submerged pump, mine, 316.
Suction, 9.
Suction-pipe, 122.
enlargements in, 99.
fire-pump, 281.
loss of energy in, 112.
Suction- and delivery-pipe, 98.
practical limitations, 110.
valves, fire-pump, 278.
Syringe, 9.

T.

Tandem compound cylinders, 236.
Tangyes's compound pump, 259.
Tank-engine, compound, 254.
economy of, 256.
Taper-taps for guard-nuts, 57.
Teague, Henry, quoted, 48.
Tensile strength, steel rods, 38.
Test of underwriter pump, 285.
Thermal units per horse-power, 412.
Throttle-valves, 3 for large pumps, 316.
Thurston, R. H., tests by, 39.
Tie-rod collars, mine-pump, 317.
flanges, mine-pump, 318.
Tobin bronze, 39, 277.
Top-guided valve, 66.
Torricelli's experiments, 10.
Triple-expansion engine, Davidson's, 264.
Reynolds's, 430.
Troy valve, 65.
Tuck's packing, 16.
Tweddell, R. H., quoted, 153.

U.

Underwriter pump, 271.
Unwin, W. C., quoted, 376, *et seq.*

V.

Vacuum-chamber, 96, 122, 281.
and strainer, 102.
in suction-pipe, 97.
Vacuum diagram, pump, 50.
Valve, annular, 64.
area, ordinary, 84.
crank-pumps, 175.
mine-pumps, 302.
ball and cage, 55.
bell, 55.
butterfly, 51.
cap, 58.
chamber lined with wood, 309.
mine-pump, 321.
clack, 47.
concentric ring, 87.
conical mitre, 53.
dash-relief, 228.
double-beat, Cornish, 70.
rubber, 70.
easy-seating, 63.
four-seated, 80.
gear, duplex, 227.
guard, hemispherical, 60.
guides, 65.
india-rubber disk, 56.
mechanically operated, 88.
metal disk, 58.
mine-pump, 319.
motions, Blake's, 204.
Cameron's, 203.
Clarkson, 207.
Davidson, 212.
Dean Brothers, 209.
Guild and Garrison, 215.
isochronal, 218.
Knowles, 201.

Valve motions, Marsh, 213.
multiple-beat, 68.
Perreaux's, 81.
size of, 84.
top-guided, 66.
Troy, the, 65.
weighted, 83.
wing, 52.
curved, 53.
with cataract, 220.
Valve-plates, 58.
Valve-seat bushings, 82.
and spindle, 57.
dovetailed, 83.
fire-pumps, 279.
inserted guard, 59.
securing in place, 61.
Valve-springs, 58.
Valves, sticking of, 279.
Velocity of flow in suction-pipe, 98.
Vertical plunger-pump, 144.

W.

Water following pump-bucket, 113.
frictional resistance, 150.
loss of energy in flowing, 112
Water, raising of, by suction, 111.
Water-elevators, 8.
Water-end design, 126.
for crank-pump, 174.
for mine-pump, 298, 317.
for trade-pumps, 13.
Water-pipe for mines, 324.
Watson and Stillman, 170.
Webber, W. O., quoted, 334, *et seq.*
Weisbach, P. J., quoted, 112.
Wing-valve, 52.
built-up, 67.
leather-face, 53.
Wolff, Ernest, tank-engine, 255.
Wood-lined air-chamber, 310.
mine-pump, 308.
Wood-packing for pistons, 20.
Woodward pump, 183.
valve, 149.
Worthington, Henry R., 226.
Charles C., 393.
high-duty engine, 393.
duty of (Unwin), 376–385.
plunger-pump, 132.
sinking-pump, 294.
tank-engine, 255.

THE END.